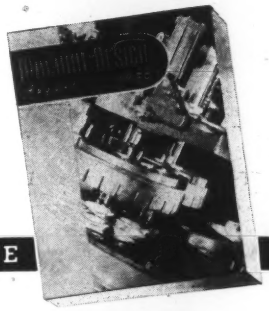

This Month's Cover: Floating rings on the Lapmaster lapping machine continuously dress the lapping plate. Designed by the Crane Packing Co., the machine produces surfaces within one light band of flatness.

MACHINE DESIGN

Vol. 22—No. 8 August, 1950

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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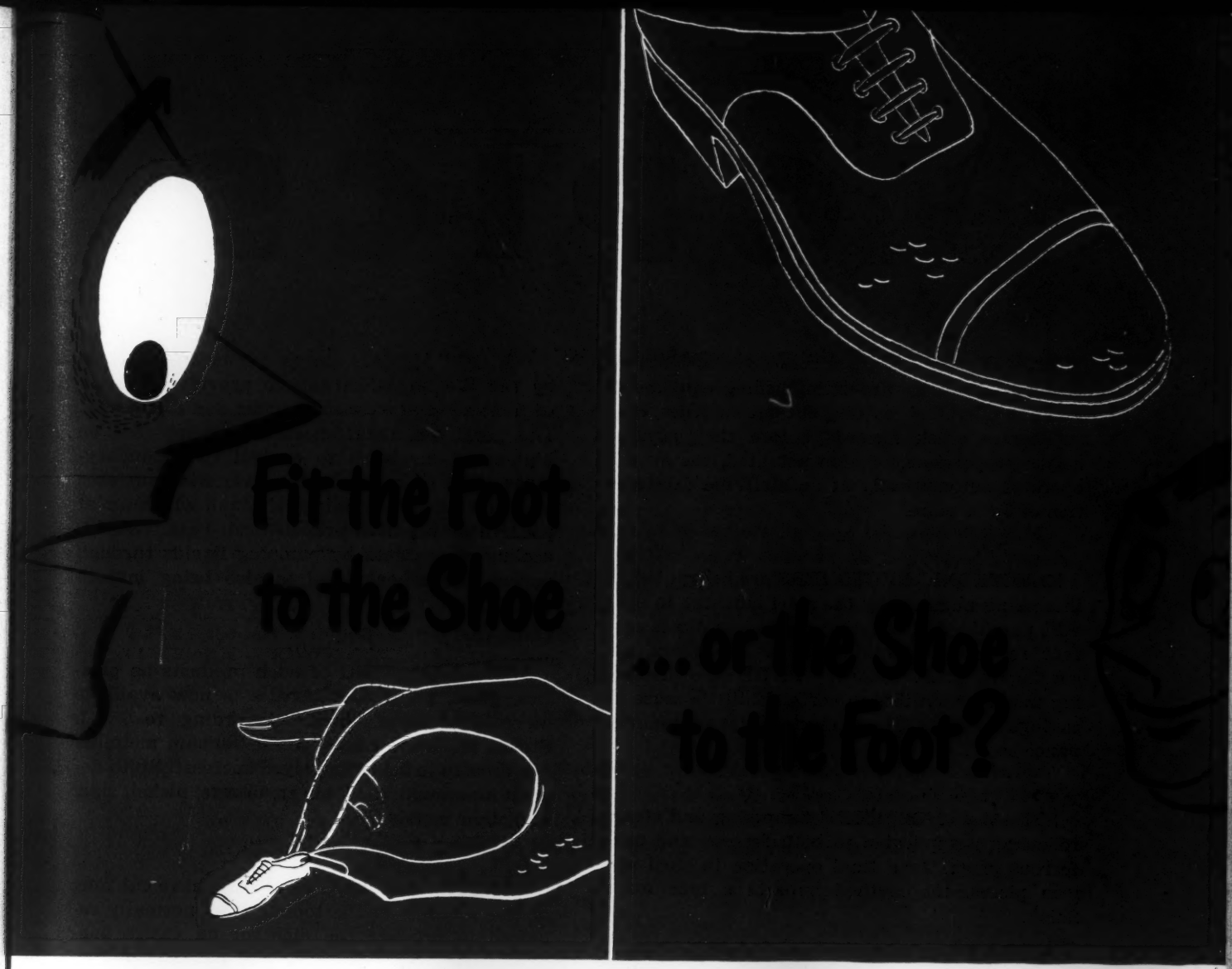

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FIRST IN FRICTION

TOPICS

CRASH SWITCH for aircraft service intended to actuate fire-extinguishing equipment, fuel and oil valves, battery cut-out switches, etc., suppresses crash hazards before they assume major proportions. Piston actuated, the switch operates automatically at an airframe deceleration of 3*g* or more.

ELECTRONIC RECTIFIERS are being used in increasing numbers by the steel industry to convert purchased alternating current to direct current required for rolling mills. Rectifiers, according to L. A. Umansky of GE, are displacing motor-generators in certain mills because of savings in installation, electricity and maintenance costs.

STEAM-ATMOSPHERE tempering and strain relieving is applicable to both ferrous and nonferrous parts. As a final operation to steel and iron pieces, the method imparts a blue oxide finish.

MAGNESIUM-ALLOY aircraft skin seven times as thick as that of the normal thin aluminum-alloy skin is being studied by the Air Materiel Command. The thick-skin plane would require fewer fittings and fastenings and dispense with internal formers and supports. Fuel space added by this change has extended the range of one model by 30 per cent.

MOLECULAR BONDING by the AL-Fin process is employed in the production of a new hobbed and shaved aluminum alloy gear with a steel hub. Lighter and stronger than molded resin and fiber gears, the new gears are claimed to be as quiet in operation as the composition type.

ATOMIC ENERGY DATA classified as nonsecret will be more easily available to American business firms under a new distribution plan announced jointly by the Atomic Energy Commission and the U. S. Department of Commerce. A clearing house for other types of nonsecret federal technical information, the Office of Technical Services of the Department of Commerce will also be the sales agency and reference source for nonsecret AEC technical reports.

POROUS METAL sheets are being produced by the Wel-Met Company to provide a variety of filtering characteristics or air and liquid flow. The powdered metal sheets are being used on high-speed airplanes to control the boundary-layer effect of rapid air flow over wing surfaces and as wind tunnel linings through which air at positive or negative pressure can pass. Sweat-cooling of surfaces by pumping liquids through porous bronze or nickel is also being investigated.

METAL COATING of such products as plastics, glass, paper and textiles is now available on a production basis. According to F. J. Stokes Machine Company, the vacuum metallizing process is being employed successfully to deposit aluminum, gold, silver, bronze, nickel, zinc, and other metals.

PLASTIC MOLDING techniques have cut four or five months off the tooling time normally required to produce the dies for an entire passenger car body. By means of plastic molding, a final plaster model for use in die-carving machines is made as a duplicate of the original clay model. Tedious, time-consuming carving of wood formerly used for the machine master has been eliminated.

STANDARDIZATION of bolt, nut and cap screw dimensions among British, Canadian and American groups is an expected sequel to the recent unification of screw thread standards. Representatives of the three countries will recommend two series: bolts, nuts and cap screws for heavy-duty machinery and a selection based, in all but one case, on present proposed American standards for either regular or light series. By unification of these details, benefits of the screw thread standardization program will be extended to include uniform clearances for wrenching.

OLD RAILROAD RAILS are transformed into many useful products without remelting, according to the American Iron and Steel Institute. Heated to rolling temperature, the rails are first slit into three parts—head, web and flange—and then rolled to the required size and shape. Concrete reinforcing bars, structural tubing and light rails are the principal products rolled.

MACHINE DESIGN
A PENTON PUBLICATION

Let's Put Them to Work!

A DISTURBING picture of the quality of America's fighting potential is painted by the distinguished military critic Hanson W. Baldwin in the July 15 issue of *The Saturday Evening Post*. In an article bearing the provocative title "We're Not the Best in the World" he concludes that, good as we were, we did not in the last war possess the best soldiers, the best generals, *or the best equipment*. With a few notable exceptions, he points out, our forces won battles only when numerically superior by a comfortable margin. He credits the successful outcome of the war to our ability to throw into the struggle, along with our allies, vast quantities of men and equipment.

What should be particularly shattering to our complacency as engineers is Mr. Baldwin's carefully documented charge that in the design of much military equipment we were woefully behind our enemies and some of our allies. Here he is referring particularly to overall design of complete military machines—guns, torpedoes, tanks, aircraft, warships—conceding that war equipment and components based on commercially developed designs were often superior to those of other nations.

Mass production being so much part and parcel of our nation's development it is entirely natural that this capacity should represent our major contribution to victory. Moreover we are not on the whole a military-minded nation and while aggressor nations were devoting a major portion of their efforts to training soldiers and building armaments we were only too eager to allow this sort of activity to become a very minor part of our peacetime economy. In other words, military design has been desultory compared with commercial design.

As the so-called cold war progresses into a shooting war with the Korean outbreak, the free world will continue to look toward the U. S. to supply the fighting forces which the United Nations will require to wage the struggle against Communist aggression. It is unlikely that we shall ever again enjoy the overwhelming superiority in numbers that helped us in the last war. Instead, our resources must be thinly spread out at the ends of extended supply lines against an enemy with interior lines. Consequently, superior quality alone can save the day.

If Mr. Baldwin's analysis is correct—and early reports from the Korean battlefield seem to confirm his findings—the job of developing powerful and efficient fighting machines cannot be left solely in the hands of the military experts. If the free world is to survive, the military experts and engineering management in industry must collaborate to bring the full talents and imagination of America's top-flight design engineers to the task. Industry has the brains; it requires only that they be put to work in the proper places.

Colin Carmichael

EDITOR

Hydraulic Pumps and Motors

New performance equations aid design of positive displacement units

FOR many years the design of positive displacement pumps and motors has been based on empirical standards and rule-of-thumb methods. Although such procedures have produced phenomenally successful hydraulic pumps, motors and transmissions, recent applications, especially in servomechanisms, have required more basic considerations than were necessary in previous days. Investigations directed toward the determination of basic facts have revealed the fundamental mechanics of the positive displacement pump and motor. It is now possible to design a pump or motor with some degree of confidence and predict its performance characteristics with reasonable assurance. There remain, however, a number of unsolved problems that require additional research to bring to light the facts necessary for the formulation of a completely sound design procedure.

A study of several designs of pumps, motors and transmissions reveals that there exists even today a rather widespread lack of understanding of the basic principles which govern the operation of pumps and motors of the positive displacement type. This can readily be seen from the claims made for new pumps and motors of radical design. There are several reasons for this situation but two are outstanding. First, there has not been available until recently any textbook devoted to the design of such mechanical devices. Second, most designers of pumps and motors are mechanical engineers who by education and experience have been concerned primarily with the mechanical and not the hydraulic aspects of machines.

Even a hasty inspection of textbooks on design of centrifugal pumps and reaction turbines reveals the essentially hydraulic character of the most important features of such design. That the hydraulic factors play an equally important role in positive displacement design is not so well recognized.

It is the purpose here to point out the hydraulic factors so important in positive displacement design and to review some of the more recent developments in this field. Basic performance equations describing

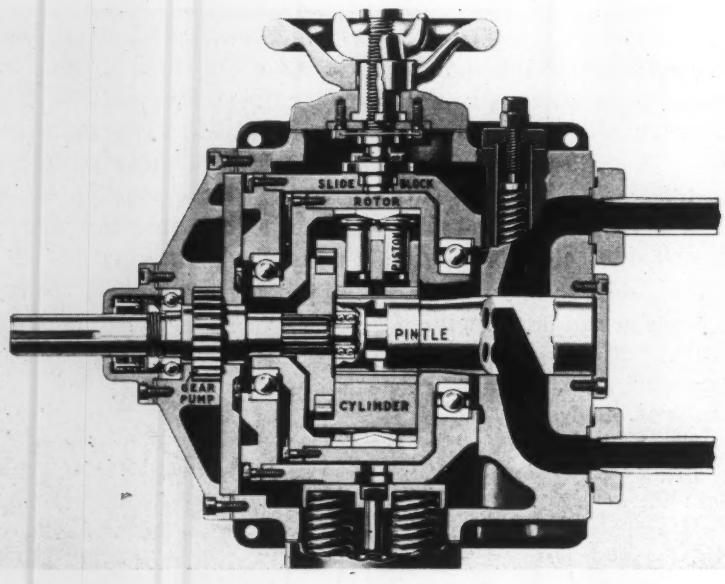


Fig. 1 — Right—Radial-piston variable - displacement pump

Fig. 2 — Center—Axial-piston pump

Fig. 3—Extreme right—Vane variable-displacement pump

By Warren E. Wilson
President
South Dakota School of Mines and Technology
Rapid City, S. Dak.

pump and motor performance will be introduced and the concepts included in these equations will be considered. Design considerations which are derived from these equations will be set forth and discussed. In conclusion, recommendations for future research will be made.

BASIC ELEMENTS: Five basic elements of a positive displacement pump or motor contribute to the hydraulic action. These are not always obvious but it will be shown that each type of pump or motor includes these five elements regardless of the type unit it may be. The five basic elements are:

1. A surface of fixed area, the positive displacement element, capable of moving through a channel of fixed cross-sectional area
2. A means of admitting liquid to the channel
3. A means of discharging liquid from the channel
4. A means of sealing the inlet from the outlet
5. A means of removing the positive displacement element from action.

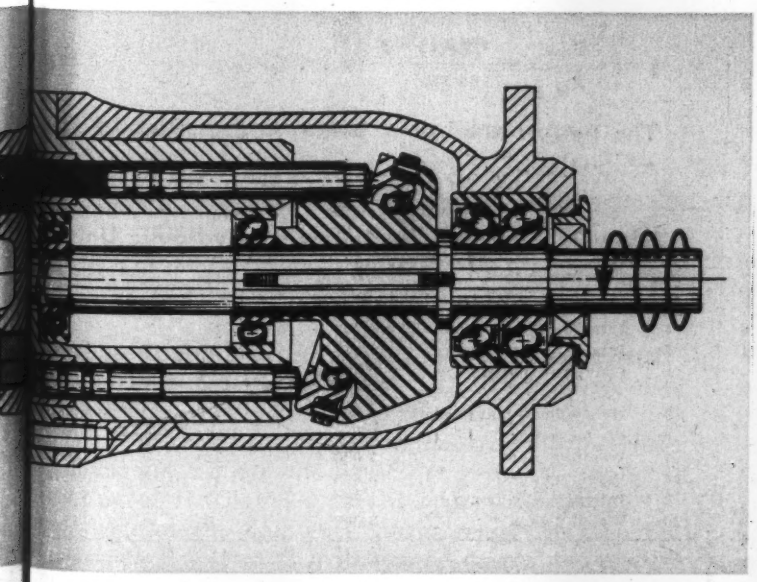
These elements can be described by citing examples in three different types of pumps; namely, the piston pump, vane pump and gear pump. These are shown in TABLE 1.

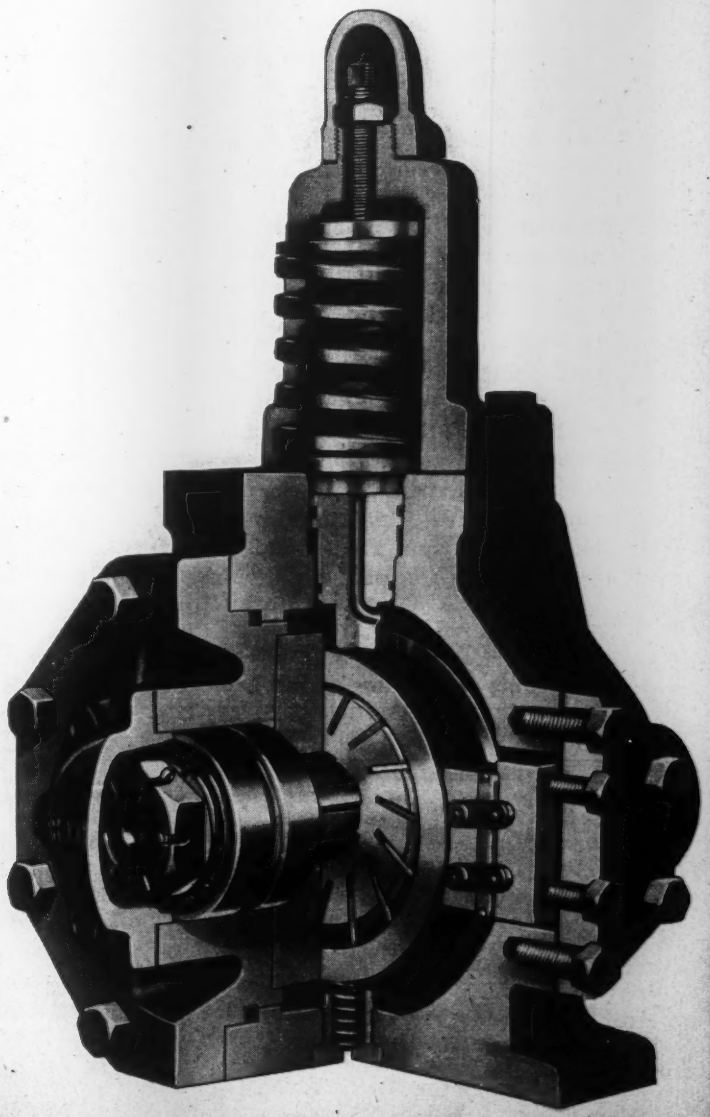
These basic elements are necessary and sufficient for the purpose of accomplishing pumping by positive displacement. The positive displacement pumping cycle from which the necessity for these elements is derived may be described as follows:

1. Liquid flows into space created by motion of positive displacement element away from inlet
2. Liquid is confined in a channel of constant cross section and transported from inlet to outlet region
3. Liquid is forced out of outlet passage by advance of positive displacement element.

This cycle and the action of the positive displacement element are reviewed briefly for each of the three types of pumps.

Piston Pump: The inlet valve opens and provides passage for liquid to enter the cylinder behind the piston as it moves in the cylinder. Pressure upstream forces the liquid into the cavity formed by the movement of the piston. When the piston reaches the end of the stroke, its motion is reversed, the inlet valve closes and the discharge valve opens. The piston advances and forces the liquid ahead of it through the outlet passage. At the end of the stroke the pis-





Nomenclature

b = Width of slip passage
C_d = Coefficient of viscous drag
C_f = Coefficient of dry friction
C_s = Coefficient of slip
D = Displacement, volume per revolution
l = Length of slip passage
n = Speed, revolutions per unit time
Δp = Difference in pressure between outlet and inlet, force per unit area
P_{in} = Power input
P_{out} = Power output
Q = Actual delivery, volume per unit time
Q_i = Ideal delivery, volume per unit time
Q_r = Loss in delivery due to cavitation, volume per unit time
Q_s = Slip flow (or delivery), volume per unit time
T = Torque
δ = Clearance between flat plates (analogy of slip passage)
η = Overall efficiency
η_v = Volumetric efficiency
μ = Dynamic viscosity of liquid

ton stops, reverses its motion and repeats the cycle. Radial and axial-piston pumps are shown in *Figs.* 1 and 2.

Vane Pump: The inlet to the vane pump is always open and provides the necessary inlet for the liquid continuously. As a vane moves away from the inlet it reaches the seal position and then forms a barrier between the inlet region and the liquid which is being forced ahead of the vane. Liquid on the inlet side will follow the retreating surface of the vane due to the pressure in the inlet passage upstream from the vane. If the speed of the vane is too great, the pressure may not be sufficient to keep the space behind the vane filled with liquid. In that case cavitation results with the formation of pockets of air or vapor. Liquid ahead of a vane is confined between two adjacent vanes after the trailing vane passes the seal point. While confined between two vanes the liquid is transported until the leading vane passes the seal point on the discharge side. The confined liquid then enters the high-pressure zone and is forced into the discharge passage by the positive action of the advancing vane. This vane is then displaced from action by the vane guides and eventually passes from the outlet region past the dividing seal into the inlet region again and the cycle is repeated. A vane pump of the variable-displacement type is illustrated in *Fig.* 3.

Gear Pump: Action of the gear teeth in a gear pump is essentially the same as the action of the vane of a vane pump and the cycle follows the same order as in the vane pump. The gear teeth are removed from action and pass through the dividing seal when they mesh. Actions of the vane and the gear pumps differ therefore only in the nature of the seal separating inlet and discharge areas and in the method of removing the teeth from positive displacement action. *Figs.* 4, 5 and 6 show various types of gear pumps.

ELEMENTARY PERFORMANCE CONCEPTS: The elementary ideas which are in vogue to describe the performance of positive displacement pumps and motors can be summarized very briefly. Power input, power output, volumetric efficiency, and overall efficiency are commonly used to describe pump and motor performance.

The power input to a pump is the product of the torque which is necessary to drive the pump and the angular speed of the shaft to which the torque is applied (see Nomenclature):

$$P_{in} = 2\pi Tn \quad (1)$$

The power output of a pump is the product of the volume rate of flow and the difference in pressure between outlet and inlet:

$$P_{out} = Q(\Delta p) \quad (2)$$

The volumetric efficiency of a pump is defined as the ratio between the actual delivery of the pump and the ideal delivery which would be indicated by the geometrical displacement of the pump:

$$\eta_v = \frac{Q}{Q_i} \quad (3)$$

The ideal delivery is in turn:

$$Q_i = Dn \quad (4)$$

The overall efficiency of a pump is simply the ratio of the power output to the power input:

$$\eta = \frac{P_{out}}{P_{in}} = \frac{Q(\Delta p)}{2\pi Tn} \quad (5)$$

The performance of a motor is described by the

Table 1—Examples of Basic Pump Elements

Element	Piston Pump	Vane Pump	Gear Pump
1	Piston in cylinder	Vane in space between rotor and housing	Gear tooth in space between housing and gear body
2	Inlet passage	Inlet passage	Inlet passage
3	Discharge passage	Discharge passage	Discharge passage
4	Valves	Dividing seal	Meshing of gears
5	Stopping and reversal of motion of piston	Retracting of vane	Meshing of gears

Table 2—Frictional Properties of Hydraulic Units

No.	Hydraulic Unit	D (cu in. per rev)	T_o (in-lb)	C_f
1	Internal-gear pump	1.205	13	0
2	Internal-gear pump	2.965	0	0.0446
3	Spur-gear pump	2.965	0	0.179
4	Vane pump	2.865	0	0.212
5	Herringbone-gear pump	0.652	4	0.200
6	Piston motor	2.060	7	0.067
7	Piston motor	1.000	12	0.094
8	Piston motor	3.365	17	0.0875
9	Internal-gear pump	1.800	13	0.075
10	Piston pump	3.600	0	0.045

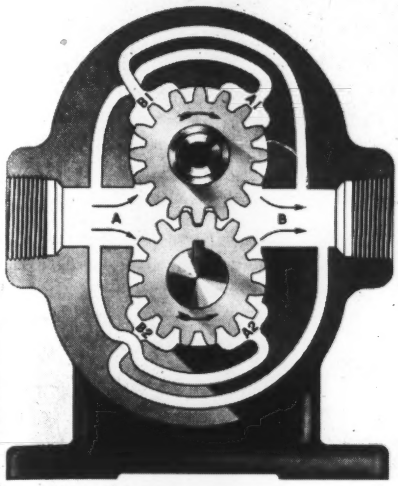


Fig. 4 — Left—Spur-gear pump

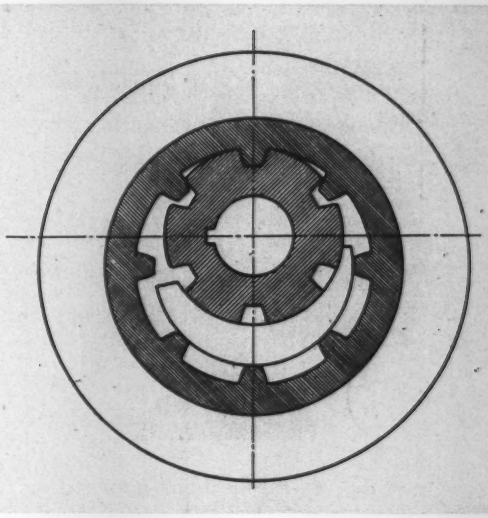


Fig. 5 — Right—Internal-gear pump

same quantities used in pump performance description but the equations defining the various elements of performance are as follows:

$$P_{in} = Q(\Delta p) \quad \ldots\ldots (6)$$

$$P_{out} = 2\pi Tn \quad \ldots\ldots (7)$$

$$\eta_v = \frac{Q_i}{Q} \quad \ldots\ldots (8)$$

$$\eta = \frac{2\pi Tn}{Q(\Delta p)} \quad \ldots\ldots (9)$$

These elementary equations sufficed for many years for the description of pump and motor performance but recent developments indicated the desirability of a more complete description of the characteristics of pump and motor performance.

NEW PERFORMANCE EQUATIONS: Performance of a positive displacement pump can be described quite adequately by the following equations (see Nomenclature):

$$Q = Dn - \frac{C_s D(\Delta p)}{2\pi\mu} - Q_r \quad \ldots\ldots (10)$$

$$T = \frac{(\Delta p)D}{2\pi} + C_d D\mu n + \frac{C_f D(\Delta p)}{2\pi} + T_c \quad \ldots\ldots (11)$$

The corresponding equations for a motor are:

$$Q = Dn + \frac{C_s D(\Delta p)}{2\pi\mu} \quad \ldots\ldots (12)$$

$$T = \frac{(\Delta p)D}{2\pi} - C_d D\mu n - \frac{C_f D(\Delta p)}{2\pi} - T_c \quad \ldots\ldots (13)$$

These equations contain sufficient terms to describe the various items that contribute to the performance characteristics of positive displacement pumps and motors, provided the clearances between moving parts and the viscosity of the liquid are such that the local heating of the liquid does not materially change the viscosity of the liquid. If such is not the case the coefficients C_s and C_d will not be constants but will show a variation with speed, viscosity and pressure differential.

INTERPRETATION OF EQUATIONS: These new performance equations contain several concepts which are of interest to the design engineer and have been employed in design studies directed toward the improvement of pump performance.

Equation 10, describing the delivery of a pump or motor, states simply that the delivery of a pump is equal to the geometrical displacement multiplied by the angular speed of the shaft less the slip flow and less the loss in delivery due to cavitation. It implies that the slip flow is due to laminar flow of the liquid through small passages within the pump. Slip flow is contained in the term $C_s D(\Delta p)/2\pi\mu$. Slip flow due to laminar flow of the liquid through small passages can be described by the equation for laminar flow between parallel flat plates:

$$Q_s = \frac{(\Delta p)b\delta^3}{12\mu l} \quad \ldots\ldots (14)$$

When the slip flows for all small passages are summed up and the total expressed in terms of the appropriate variables, the quantities $\Delta p/12\mu$ are common to each term of the sum and can be factored out of the series. The various terms of the series being summed, of the form $b\delta^3/l$, have the dimension length to the third power. The geometrical displacement D also has the dimension length cubed. Hence by the reasoning that displacement is related to geometrical characteristics of the pump in roughly the same manner as the width and length of slip path and clearnace, the term $b\delta^3/l$ is replaced by $C_s D$. The coefficient C_s then becomes an index of the geometrical features of the pump that contribute to slip.

Equation 11, for torque, describes the torque in a manner similar to that used in the description of delivery. The ideal torque required to drive a pump is said to be the torque required to rotate the shaft when the pressure differential is Δp, namely, $(\Delta p)D/2\pi$. In addition, the viscous drag developed between a moving and a stationary surface with small clearance causes a retarding torque of magnitude, $C_d D\mu n$. This term is of the form necessary to describe

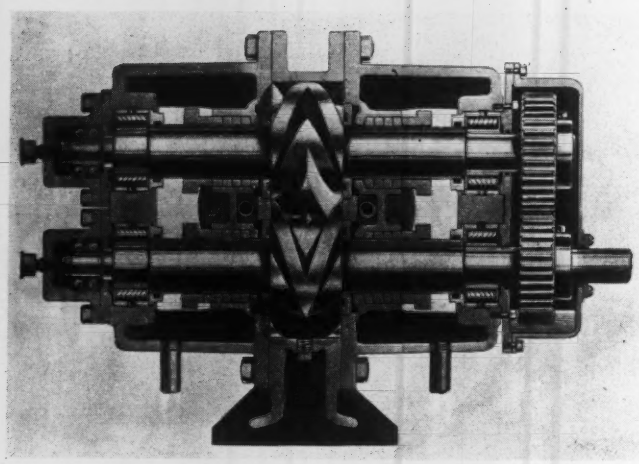


Fig. 6—Herringbone-gear pump

viscous torque, drag being proportional to the relative speed of the moving surfaces which in turn must be proportional to the angular speed of the shaft. The drag is also proportional to the viscosity of the liquid between the moving surface and the stationary one, proportional to the area of the surface and inversely proportional to the distance between the surfaces. The use of the quantity C_dD to replace a term consisting of a summation of terms containing the quantities b, l, δ and the radius of the pump rotor is justified in the same manner as the use of the term C_sD in describing slip flow.

Retarding torque $C_fD(\Delta p)/2\pi$ is a dry friction torque originating at some location such as a journal bearing or other supporting element where the drag is proportional to the supported load which in turn is proportional to the pressure differential.

Torque loss T_c should not exist in a well-designed pump because it represents a torque caused by a force generated at metal-to-metal contact which is unlubricated and is not affected by the load due to pressure differential. Such torque can be caused by a rough spot on a moving surface rubbing a stationary surface.

The concept of torque efficiency, introduced recently, is useful in evaluating pump and motor designs. In a pump it is the ratio of ideal torque to actual torque and in a motor it is the ratio of actual torque output to ideal torque. The product of torque efficiency and volumetric efficiency is equal to overall efficiency.

The designer should have well in mind these concepts:

1. Slip flow increases with the cube of the clearance
2. Viscous drag is inversely proportional to the clearance.

Knowing these facts, the designer can control the slip and the drag by the clearances that he plans into the pump. However, he cannot make both slip and drag arbitrarily small. If he decreases the clearance to reduce the slip he inevitably increases the viscous drag. If he increases the clearance to decrease the drag he materially increases the slip.

It is possible to design any given slip passage for a minimum power loss in that passage and to design a pump so that the power loss is a minimum for any given set of operating conditions. Such a design would have clearances so proportioned that the sum of the power losses due to slip and to viscous drag would be a minimum. Under these circumstances the pump would operate at maximum efficiency at the planned operating conditions of speed, pressure and liquid viscosity.

Although this design procedure is simple and effective, it is based unfortunately on one assumption which is not realized in practice with existing hydraulic liquids or, in fact, with any liquid. This assumption is that the liquid viscosity is constant. This is, of course, not true if there is any local heating of the liquid in the small passages through which the slip occurs and in which the viscous drag is of considerable magnitude. Current research on the evaluation of the effect of this local heating will, it is hoped, result in practical design techniques that will make possible an optimum design for any given set of operating conditions.

All Design Elements Important

To illustrate the significance of the equations which describe pump performance and the necessity for giving consideration to all elements of the design which determine the performance, *Fig.* 7 is useful. In this illustration the power input to a pump is shown as a function of pressure at a particular speed and with a single liquid. The various items that absorb this power input and the power remaining as power output are indicated.

The top line indicates the power input at various pressures for a gear pump operated at a speed of 1500 rpm with a liquid of approximately 300 *SSU* viscosity at the inlet. The vertically cross-hatched area indicates the power absorption due to viscous friction is essentially constant with increasing pressure; it varies with speed but not with pressure.

The diagonally ruled area also represents a constant power loss which is due to a dry friction unaffected by the pressure differential.

The area with horizontal lines represents the power loss due to slip. This increases rapidly with pressure since the slip itself is directly proportional to the pressure and the power loss is equal to the product of the pressure and the slip giving a second-power increase of power loss with pressure.

When operating at full design pressure, 1000 psi, and at 1500 rpm, this pump shows a power input of 5.2 hp and an output of 3 hp indicating an efficiency of 58 per cent. The slip accounts for a power loss of 1.6 hp or 31 per cent of the input power. Viscous drag and dry friction each account for approximately 0.3 hp or approximately 5½ per cent of the input power.

At 500 psi the distribution of power is quite different. Input is 2.9 hp, output 2.0 hp, and efficiency 69 per cent. Total loss is 0.9 hp, 31 per cent, and is comprised of approximately 0.3 hp, 10 per cent, for each of the three causes: viscous drag, dry friction and slip.

A simple study of the performance of this pump to determine only overall efficiency and volumetric

efficiency would not be sufficient to track down the design changes which would be necessary for improved performance. On the other hand a detailed study, which determined the distribution of power losses as illustrated in *Fig.* 7, gives the clues required for such a study. Location of the point of peak performance can be changed and efforts can be made to reduce either viscous drag or slip. The dry friction torque probably orginated largely in the journal bearings. Use of roller bearings would reduce this loss but add to the cost. Studies of the desirability of such a change can be made easily.

GENERAL DESIGN PRINCIPLES: Study of the performance equations has revealed a number of significant criteria and facts useful in design of positive displacement pumps and motors. As indicated previously, the elements of a pump may be designed to give peak efficiency under specified operating conditions. A second principle is that there is no gain in building a very complex mechanism. A simple piston or vane contains all the requisites of a wholly satisfactory pumping element. The simplest possible auxiliary devices to effect the activation and inactivation of a piston or vane then necessarily give the best mechanical performance.

Five elements of a pump or motor determine its performance:

1. Size of the positive displacement element (piston, vane or gear tooth) and length of its path of motion determine the displacement
2. Size and proportions of the inlet passages determine the cavitation losses
3. Size and proportions of the discharge passages determine fluid friction losses at the discharge end
4. Seal between inlet and outlet is a major factor in determining the magnitude of the slip. Clearances around the positive displacement element determine an additional part of the slip
5. Mechanical features connected with actuating the vanes or guiding the pistons or with the form of gear teeth determine to a considerable extent frictional losses of the dry friction or thin-film lubrication type.

Commonly accepted notions indicate a greater concern with leakage causing slip than with frictional losses. This is clearly shown by a study of the dry friction coefficient C_f and the constant torque loss T_c associated with a number of pumps which have been tested. In TABLE 2 are shown typical values of the quantities determined in an extensive series of experiments under the author's direction.

There is no justification for values of T_c of the order of several inch pounds or of C_f in excess of 0.03. Devastating effect of these excess values is easily seen by considering unit No. 9 in TABLE 2. The value of C_s is 1.6×10^{-7} and C_d 16.8×10^{4}. These together would permit a maximum efficiency of 84 per cent providing T_c and C_f were zero. However, with a value of C_f equal to 0.075, as for this pump, the maximum efficiency possible even with T_c equal to zero would be about 78 per cent. With T_c equal to 13 in-lb the maximum efficiency would be even less. The magnitude would depend upon the conditions of operation since maximum efficiency is not

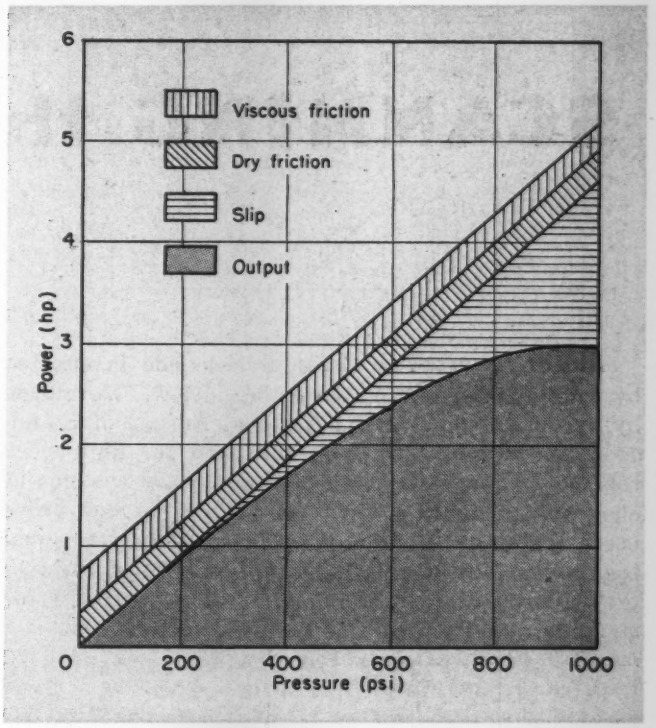


Fig. 7—Losses and output for a gear pump operating at 1500 rpm with a liquid of approximately 300 SSU

independent of the pressure, speed and viscosity when T_c is not zero.

Design of positive displacement pumps for pumping liquids of relatively low viscosities has progressed far, simply by the time honored method of cut and try. On the other hand there appears to be a fruitful field for investigation in the design of pumps for the handling of extremely viscous liquids. The theory indicates that if large clearances are used, higher speeds than those commonly accepted in commercial operation would be possible with even higher efficiencies than are now expected.

FIELDS FOR FUTURE RESEARCH: Determination of the effects of local heating on pump and motor performance and the formulation of a design procedure to include these effects is of prime importance.

Determination of the optimum gear tooth form from the standpoint of viscous and dry friction forces at the meshing and flow in the neighborhood of the inlet is essential.

Determination of the causes of noise in pumps and motors presents a challenging field since many pumps are far noisier than is desirable after all reasonable and readily determinable causes of noise have been eliminated.

Correction

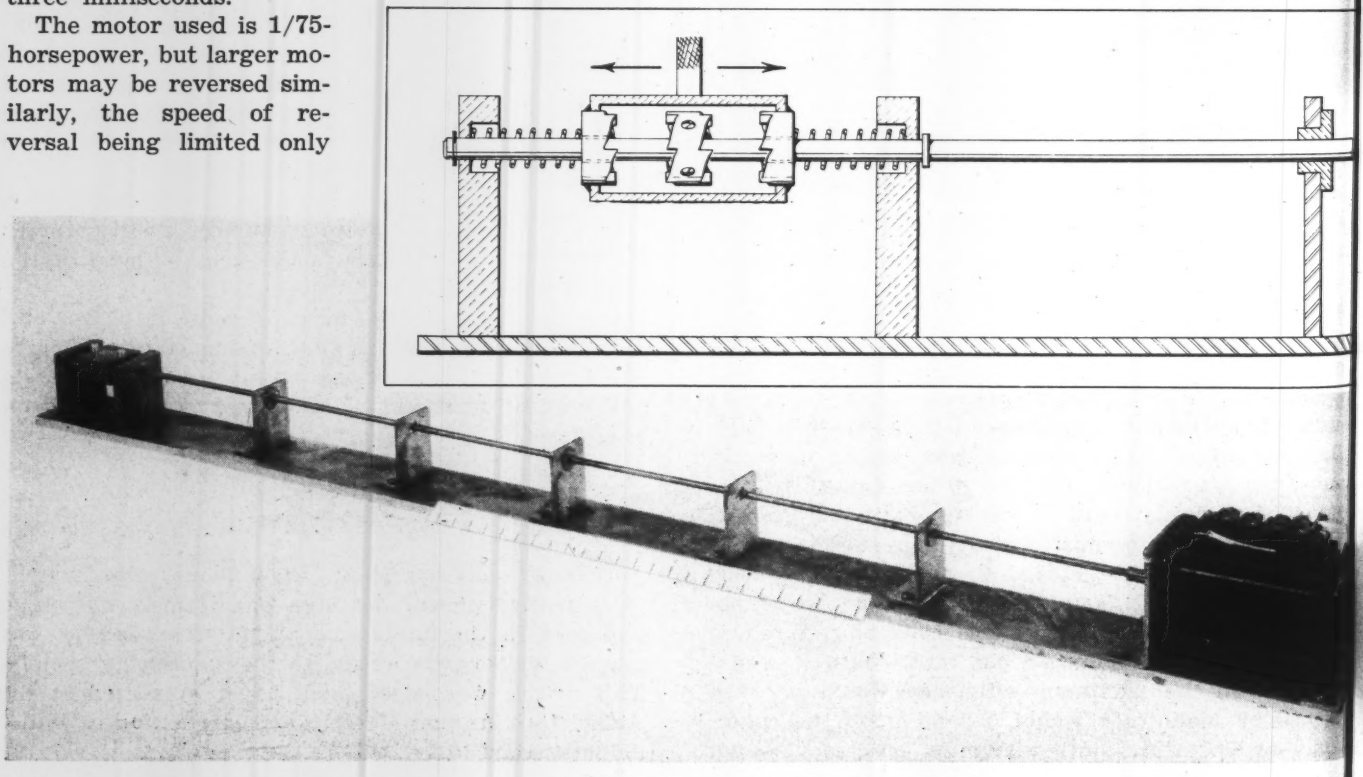
Through an oversight in makeup, the article, "Design Economics," by John Van Hamersveld which appeared in the July issue of MACHINE DESIGN was printed without the credit to the sponsoring society. This article was based on a paper presented at the ASTE 18th Annual Meeting and Exposition in Philadelphia, April 10-14, 1950.

SCANNING the Field For Ideas

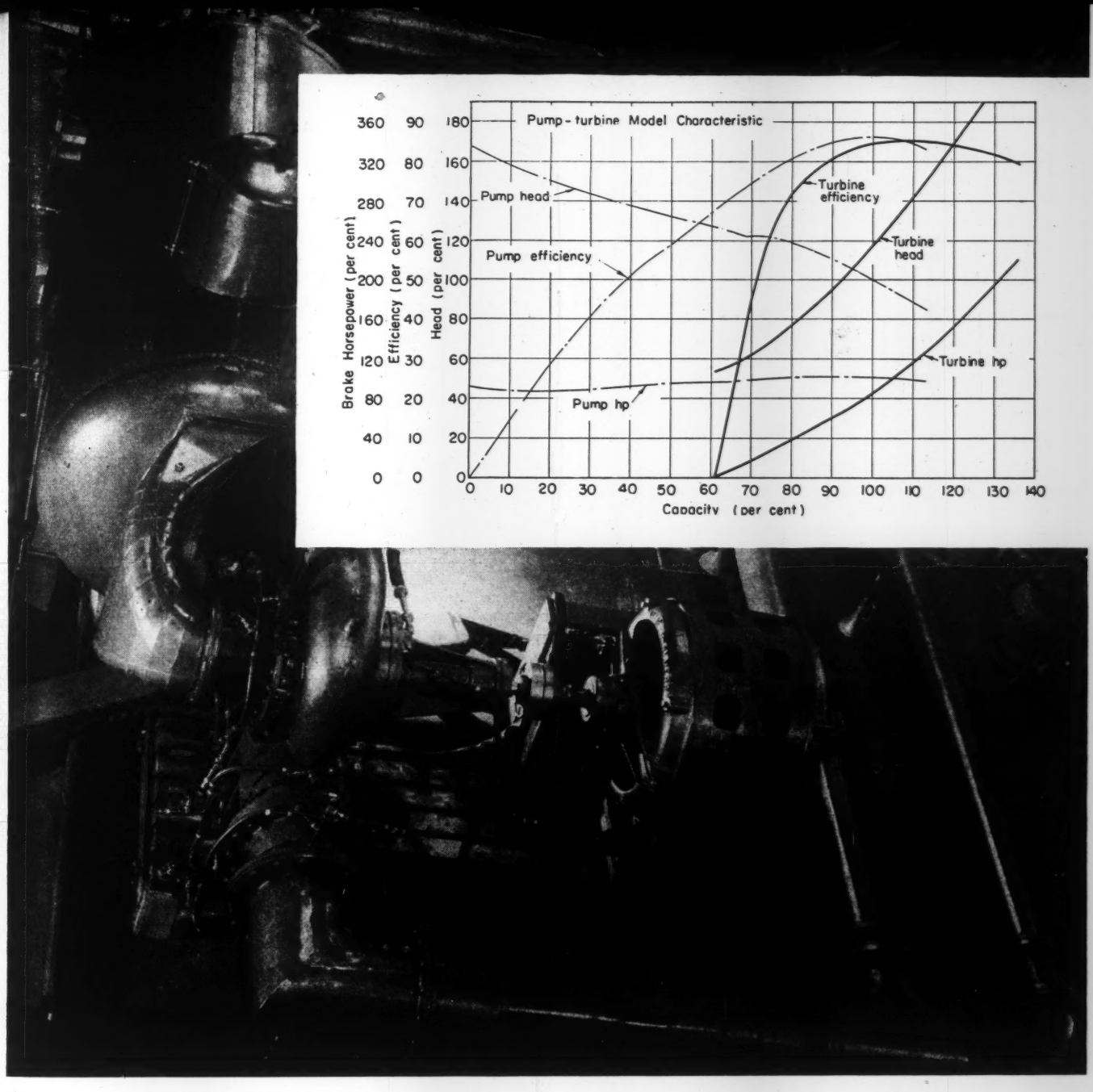
Motor reversal in three milliseconds is effected by a torsion bar and clutch device, below. Developed by Jacob Rabinow at the National Bureau of Standards and designed to meet the need for high-speed reversal of magnetic tapes in the memory systems of electronic computing machines, the device may prove useful in many other applications. During reversal the torsion bar stores the kinetic energy of the motor's rotor and then uses it to accelerate the rotor rapidly in the opposite direction. The motor is a small low-inertia two-phase unit operating at 3200 rpm. The torsion bar is 3/16-inch diameter and approximately 31 inches long. Only one phase of the motor is connected to the power supply, allowing the motor to operate in either direction of rotation.

One end of the torsion bar is connected to the motor shaft, the other is equipped with two positive unidirectional clutches as illustrated in the diagram. One clutch prevents clockwise and the other counterclockwise rotation. If the motor is running in a clockwise direction and the proper clutch is engaged, the end of the torsion bar is stopped, bringing the motor to a stop in approximately 20 degrees. The potential energy stored in the bar is then returned to the rotor as a counterclockwise impulse. The motor attains reversed full speed within three milliseconds.

The motor used is 1/75-horsepower, but larger motors may be reversed similarly, the speed of reversal being limited only by the mechanical strength of the system. Because, with this method of reversal, energy taken from the system during the stopping phase is returned to accelerate the reversed direction of rotation, heating of the motor and current surges through it are prevented. Other types of energy-storage devices could be employed similarly such as coil springs or pneumatic cylinders. Also, a stationary storage system could be used, locating the reversing clutches between it and the motor shaft.



Pump-turbine, having a single runner enclosed in a casing, serves as a pump when driven in one direction of rotation and as a turbine for driving in

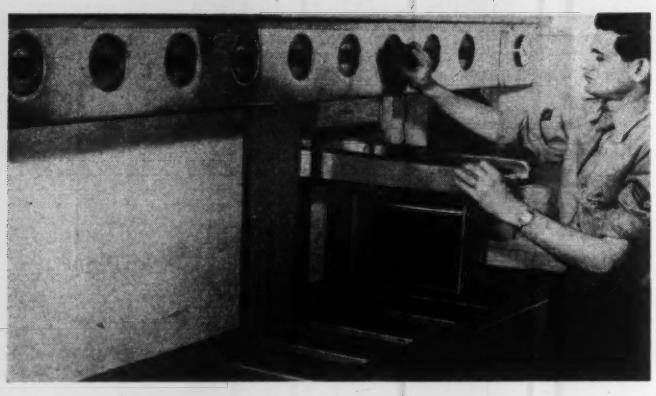


the other direction. A synchronous unit with the necessary control equipment is direct connected to the pump-turbine, acting as a motor to drive the pump and as a generator to supply power when driven in the opposite direction by the turbine. Shown above, on test at Allis-Chalmers, this machine has demonstrated that high efficiencies are attainable with a dual-purpose unit and has revived the old idea of pumped-storage hydroelectric power.

Off-peak pumping from a low-level to a high-level water reservoir and peak-period power generation with reverse flow are economically feasible with a pump-turbine combination unit. Need for additional pumping equipment is eliminated, allowing the efficient utilization of equipment for 24 hours a day. In the curves are shown the characteristics of the pump-turbine for both pumping and driving cycles. For these curves the pump head, capacity and horsepower at best point of efficiency for 100 per cent gate were used to calculate the percentages.

Pumped storage hydroelectric systems would require electrical equipment of the same general nature as is used in generating stations. Although more electrical equipment would be needed, some features such as voltage regulation and pilot excitation might actually be eliminated. Chief differences arise from the necessity of starting and operating the synchronous machine as a motor, with the direction of rotation opposite to that required when it operates as a generator. Allis-Chalmers engineers recommend the use of compressed air to start the pump unwatered to bring the drive to synchronous speed.

Automatic photo processing of color transparencies is provided by a novel machine which also permits the operator to preset individual timers for each chemical bath in the process. Each film going through the machine receives uniform treatment unless the controls are varied. In use at Air Materiel

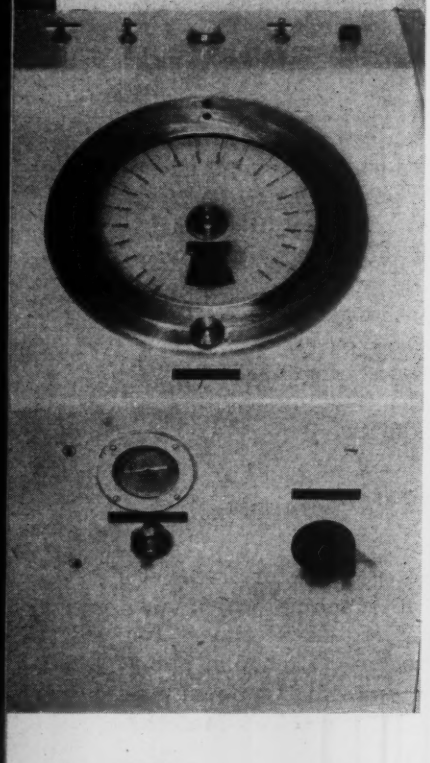


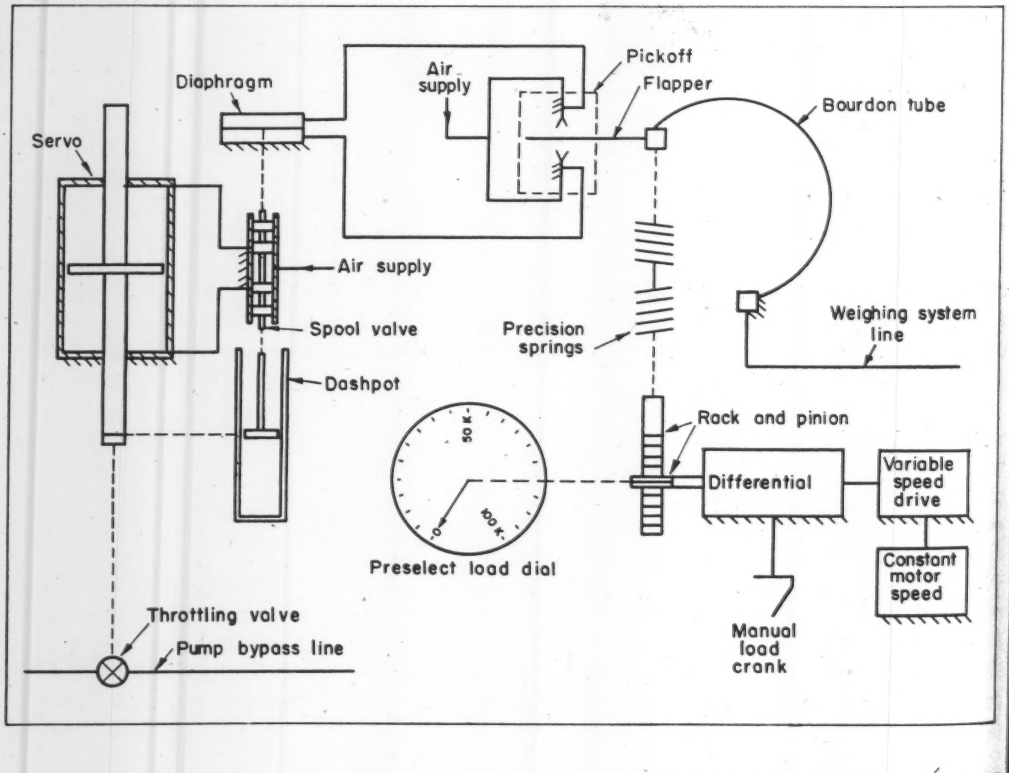
Command Headquarters at Dayton, Ohio and designed by the Newton-Niles Co., the machine above controls the processing temperature at 68 F. Wash tanks in the machine are equipped with jets to provide a constant automatic flow of clean water during the immersion time. The machine weighs 1100 pounds, has stainless steel tanks and processes transparencies in 90 minutes.

Controlled loading of a test specimen in a conventional hydraulic testing machine is obtained with the unit illustrated at left below. Developed at the Langley Aeronautical Laboratory by Rupert S. Rock, H. D. Garner and E. D. Lounsberry, the controller requires no attention from the operator after the initial settings have been made. Within the elastic limits of a specimen, the instrument has been designed to maintain constant loading with an accuracy of plus or minus 0.1 per cent of full scale reading, maintain a constant rate of loading with an accuracy equal to that of a human operator, and perform the transition from constant rate to constant load condition automatically with not more than 0.1 per cent overshoot.

Null balance principles were utilized in the design of the controller, the elements of which are indicated in the diagram, right below. A manual load crank varies the tension of a pair of precision springs through a rack and pinion assembly. These springs restrain the tip travel of a Bourdon tube connected in the weighing system of the testing machine. A preselect load dial, geared to the rack and pinion assembly, provides a calibrated indication of load which, in turn, is proportional to the tension of the precision springs. The tip travel of the Bourdon tube is translated into a pneumatic signal by a high sensitivity pickoff of the opposed venturi jet and flapper type. This signal actuates a power servo through a diaphragm operated spool valve to control the throttling valve. The throttling valve, located in the pump bypass line, is a needle plug type designed to load linearly with valve movement.

When the load is increased by rotation of the manual crank, tension on the springs attached to the Bourdon tube is increased. The resultant displacement of the Bourdon tip moves the flapper which changes the air signal from the pneumatic pickoff, unbalancing the air pressure on the diaphragm. Motion of the diaphragm and spool valve actuates the servo to position the throttling valve, increasing the load on the specimen and also the pressure in the Bourdon tube in the weighing system line. When the desired position is reached the flapper will be returned to the null position. Application of load at various rates is obtained by increasing the tension of the springs through a constant-speed motor and variable-speed drive, stopping the motor with a limit switch when the desired load is reached.





Engineering Management—Its Job

The whys and wherefores of engineering management, how it is organized and carried out from the inception of an idea to the finished machine or product

By Randolph W. Chaffee
Consulting Engineer
Cleveland, Ohio

" . . . careful research and investigation . . . to focus engineering resources in the best direction . . . "

Photo, courtesy General Electric Co.

Part 3—Planning of Engineering

NEED for management as applied specifically to the practice of engineering has been recognized but recently. At a previous time when one individual or a very small and well-knit group conceived, carried out and completed the typical engineering project, the need for literal management of engineering did not exist. Engineering was an isolated undertaking which neither influenced nor was influenced by the other functions of business, industry and commerce. Engineering was largely the personal concern and risk of the individual engineer who was his own board of directors, president, general manager, sales manager, treasurer and all other officers of business, and he answered only to himself.

A great deal of engineering, indeed probably the greatest portion, is still a matter of individual effort and susceptible only to management by the individual or small group concerned. However, the trend of individual creative engineering is downward while the scope of corporate engineering is broadening. Corporate management demands the highest of per-

formance, whether in sales, production or engineering, and rightly so. Only by such performance are capital and personnel resources employed to the maximum advantage of the enterprise, its owners, customers and employees. The need is growing for explicit management of engineering and the engineering profession is wisely seeking principles and practical methods for managing engineering.

PROBLEMS OF ENGINEERING PRACTICE: Apart from the technology of creative engineering, its practice generates numerous problems that must be recognized in the process of putting together a code for engineering management.

Engineering Employment Security: Upon being asked his opinion of engineering, the typical industrial executive will assert the vital importance of engineering to American industry, to the American people, and the tremendous contribution of engineering to the world dominance of American industry. However, faced with a declining market, disappearing profits and a need to retrench, that same industrial executive will lay off his engineering staff among the first—and just at the time when he should use the engineering staff to combat competition and to attract the remaining buying power to his product. Why does he do this and why is the engineers' employment status correspondingly precarious? Apparently it is not because of the executive's lack of confidence in engineering in the abstract, but because of his lack of confidence in the practice and conduct of engineering with respect to its ability to rise to meet the emergency situation. If engineering management has any purpose, it is to enable engineers to win the whole confidence of industry to the point where engineering is regarded as the most valuable asset of industry and the resource never to be relinquished.

Personality in Engineering: The practice of engineering is a highly individualized undertaking in which the individual is the source of original creative thinking. While highly practical in his technical approach, the engineer is inherently a subjective and emotional personality in much the same manner as the artist. As a result, the typical engineer may be introspective and not fully aware of the extraneous influences which affect his work and which his work affects. One purpose of engineering management is to bring the engineer into intimate contact with these extraneous influences so that he is made aware of them and takes them into consideration during the conduct of his work.

To illustrate, a given engineering project will have some more-or-less definite value to the enterprise which sponsors it. The redesign of a mousetrap, to make it more efficient, more attractive and easier to sell, may be worth about $15,000 to the mousetrap manufacturer and a larger expenditure cannot be recovered through increased sales. An engineer is assigned to redesign this mousetrap.

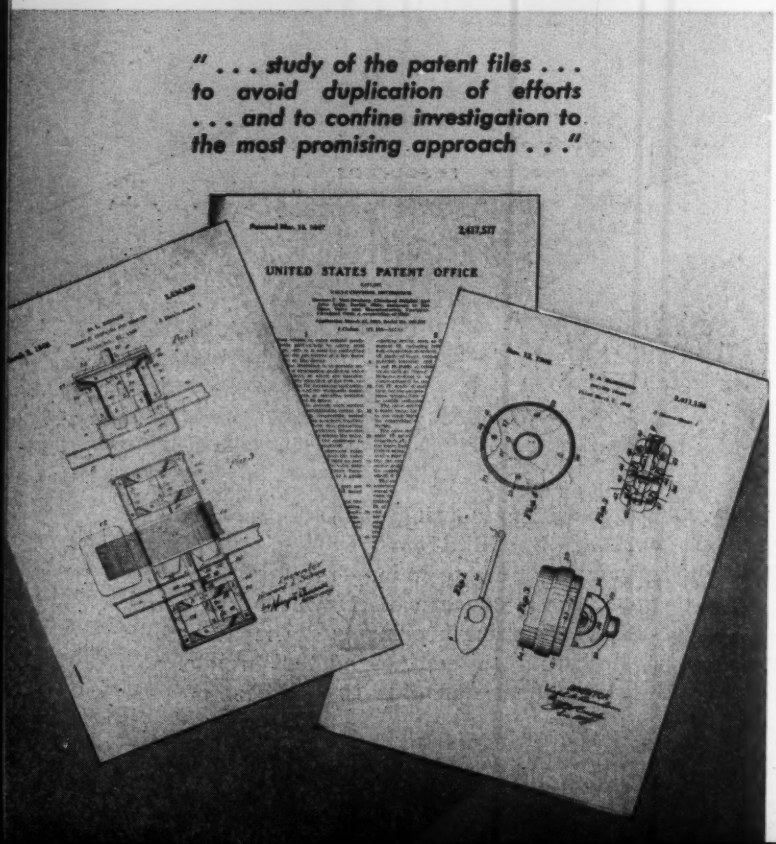
Practical Objective Necessary

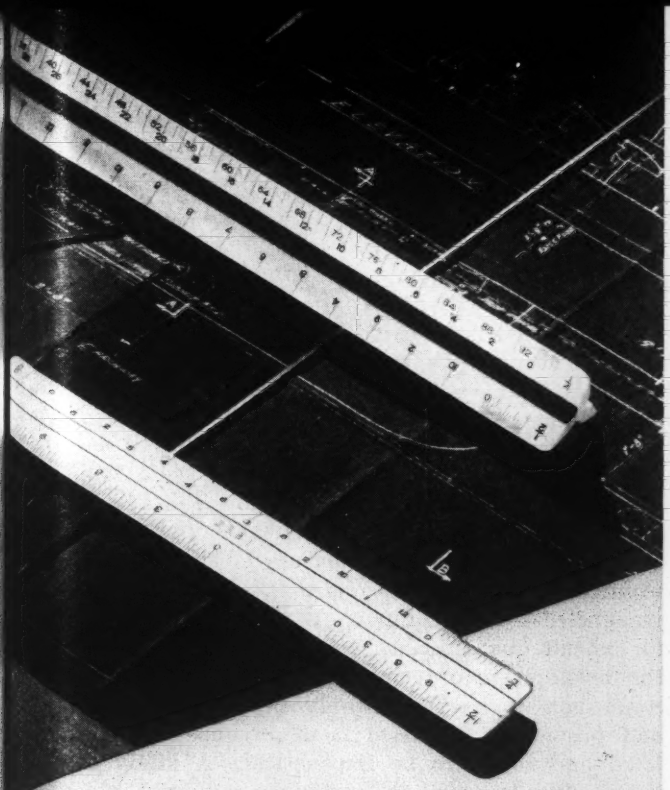
At the outset, the engineer conceives a certain objective which when reached, will mark the completion of his project. As he advances into the project, however, and learns more about mousetraps, the engineer refines his objective and sees a more-and-more perfect design of mousetrap. This refinement of objective moves the horizon farther and farther away, entices the engineer on to ever better effort and achievement, and postpones the completion of the project. The project may, if allowed to run unchecked, expend far more than the allowable $15,000 and thus defeat the whole purpose of engineering.

Co-operation and co-ordination are too often lacking in engineering. With several groups at work on a common project and each group intent upon its part, there is a hazard that each group will minimize or ignore the work of other groups, that the project components will diverge from each other, that the common goal will be lost from sight, and that capital and personnel resources will be wasted and the project delayed. A second purpose of engineering management is to enforce a proper degree of co-ordination and co-operation between engineers as individuals and in groups.

A decision to undertake an engineering project can result in an immediate putting of pencil to paper on the drafting board and putting of spade to earth in the rush to complete the project. This inaugurates a wasteful trial-and-effort approach that can be avoided by careful research and investigation to eliminate the unpromising approaches and to focus engineering resources in the best direction.

Many engineering projects have been undertaken, in the past, with an inadequate or unreasonable forecast of the time, manpower and money to be required to complete them. While these unrealistic forecasts may have been the product of the best guess that could be made, the fact remains that failure to complete projects within the predicted time and cost has weakened the status of engineering and of the engi-


" . . . study of the patent files . . . to avoid duplication of efforts . . . and to confine investigation to the most promising approach . . ."
UNITED STATES PATENT OFFICE



Photo, courtesy Eastman Kodak Co.

" . . . pencil is put to paper on the drafting board and the end product first begins to take composite form . . ."

neering profession. Engineering management has a prime and vital purpose to perfect the planning and budgeting of engineering projects and to provide the means for relating actual performance to plans and budgets.

The engineering personality is self-motivated but resents immediate supervision of its behavior. While in certain cases such as the military environment, engineers can be commanded as to conduct and behavior, this device is not generally applicable in industry. Engineers do respond, however, to intelligent and sincere aid and support. Given the right tools and information, they will police their own conduct far more effectively than any outside influence. A purpose of engineering management is to provide engineers with the right tools and information so that they may carry on their work in proper relationship and balance.

The foregoing are but a few of the typical problems arising out of the impact of personality upon engineering practice. One can recall many others of equal or greater significance.

Two Major Parts To Management

PLANNING & DIRECTION: The management of any undertaking involves but two major parts. One of these is *planning* which occurs before the project is actually started. The other is *direction* which occurs after the project is started and which continues through its duration.

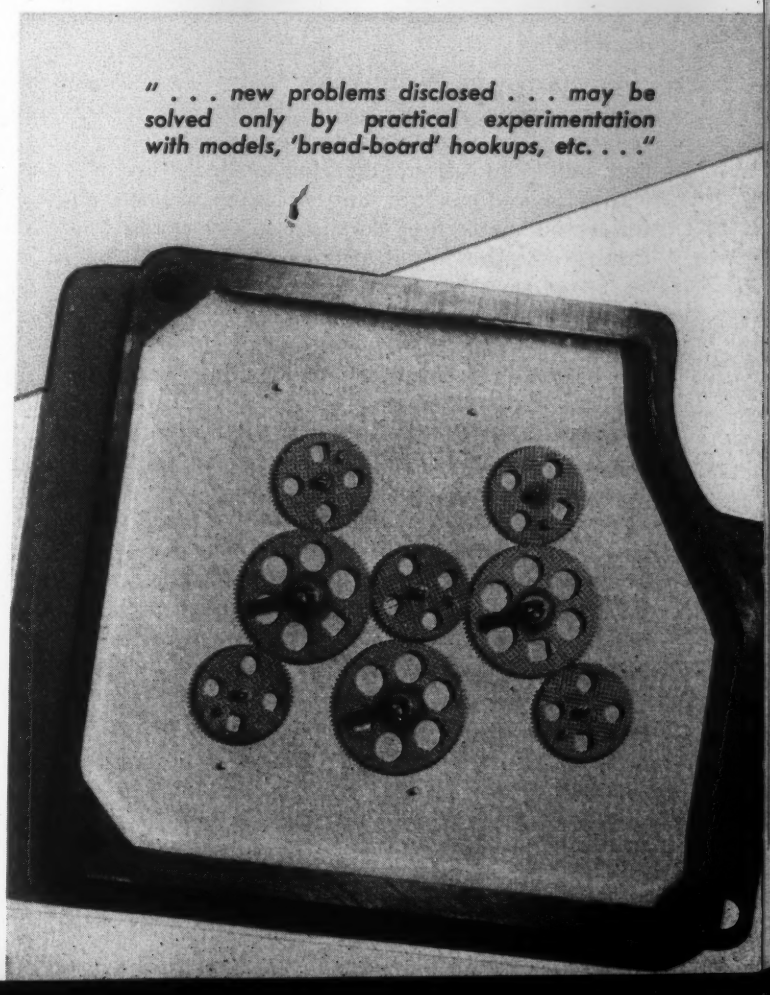
Planning and direction together make up 100 per cent of engineering management. With planning amounting to 10 per cent, direction will be 90 per cent whereas if planning uses up 90 per cent, direction requires only 10 per cent. To date, the practice of engineering has been marked by a minimum of planning in the specific sense. This is not to say, however, that a minimum of planning has been more advantageous or is to be recommended for future practice. With least planning, direction becomes more difficult, more awkward and more costly. With only a limited exploration of the problem during planning, and concerted attack upon a single avenue of approach, direction follows the trial-and-error method and must attack several approaches before the correct one is found.

With adequate planning, direction is easiest for the reason that an accurate and complete plan is virtually self-directing. Engineering resources are concentrated upon a single approach and a minimum expenditure of these resources brings the engineering to its proper conclusion in least time and at least cost.

Admittedly, the art of advance planning the conduct of a project of creative engineering is very young and as yet little experience has been had with it. Also, engineering planning is not easy because it deals with unknowns and unforeseeables and because the things learned today govern what is done tomorrow. We cannot literally look ahead to forecast what we will learn six months hence and what we shall do thereafter as a result of that learning.

Planning is nothing more than the employment of

" . . . new problems disclosed . . . may be solved only by practical experimentation with models, 'bread-board' hookups, etc. . . ."



Photo, courtesy Westinghouse Electric Corp.

" . . . testing models . . . appraising test results . . . interpreting test results into practical design recommendations . . . "

experience, and the fruits of experience are available only in the record. Therefore, experience in engineering practice must be recorded in vital detail so that the capacity to plan intelligently and realistically is created, and up to a point, that capacity will develop as the record of experience develops. The management of engineering has the purpose to create, employ and perfect the record of engineering practice to the end of more exact and perfect planning for engineering.

PLANNING FOR ENGINEERING: If many different engineers or groups of engineers were to analyze the records of completed engineering projects, they would find, not that all engineering projects are alike in character and conduct, but that each project followed its certain pattern of conduct. They would find also that, although the patterns of conduct differed widely, all of these patterns can be fitted into an overall pattern of basic principle for planning purposes. When expressed in broad terms, the overall pattern can be made to cover at once engineering projects in the various fields of mechanical, electrical, hydraulic, civil, industrial, chemical, production, and other branches of engineering, in fourteen all-inclusive and successive stages.

Stage A—Preliminary Investigations: Although not completely formalized in all cases, engineering projects start with some sort of preliminary investigations to clarify the nature of the problem, and to outline the realm within which the solution will be found. In the field of mechanical engineering, for example, preliminary investigations must define the technology, function, structure, appearance, and performance of the equipment. Parallel units are studied in respect to their virtues and deficiencies, sales experience, profit probabilities, price structure, channels of distribution, and opportunities for improving upon them. Where necessary, impartial market studies and analyses are made to assess the current market requirements and to set up the specifications for the machinery which gives most promise of widespread market acceptance.

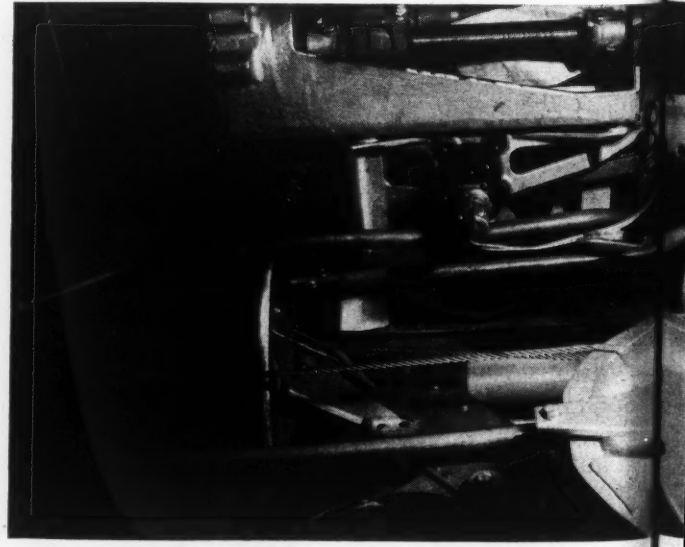
Whatever the branch of engineering involved, the purpose of the "Preliminary Investigations" stage is to survey the project from the point-of-view of the end user of the product and to visualize his requirements. This end-user may be the housewife who will use the vacuum cleaner, the assembler who will use the spot welder, the child who will use the playground equipment, or the doctor who will use the hypodermic needle. And requirements must be fulfilled if the product is to be saleable.

Stage B—Studies of Prior Art: The oft-repeated saying, "nothing is new under the sun" must also be recognized by the engineer. It would be most foolish to expend quantities of time, energy and money in the development of a device, process or method which could not be used profitably because someone else had previously patented it or because it had been tried and found to be impractical. A study of the patent files enables the engineer not only to avoid infringement on protected art but, what is more important, to avoid duplication of efforts which have been expended unsuccessfully and to confine the investigation to the most promising engineering approach.

Record of Prior Art Important

Almost without exception, the study of prior art is an essential stage in creative engineering and includes, along with patents, the record in technical or trade journals, sales literature, and engineering papers presented before the engineering societies.

Stage C—Calculations and Specifications: Using the information obtained during preliminary investigations and studies of prior art, calculations are made and specifications drawn up to guide the creative development effort. These calculations and specifications are, in effect, the guideposts pointing the way along which the engineering project will travel,



avoiding fruitless and wasteful detours, and moving in a straight line to the target. We deal here with the general characteristics of the end product of engineering, and seek the answers to such questions as

What will the product do?
When will it be used?
Who will use it?
Why is this product being created?
Where will the product be used?
How much is it worth?

and a host of others.

With the answers to these questions, we next make calculations to determine, among other things

What will the product be made of?
How big will it be?
How strong must it be?
How fast will it operate?
How much work must it do?
How much power will it need?
What will be its efficiency?

These calculations are made before there is any real vision of the end product and they tend to enforce the objective, impartial and wholly practical point-of-view. They tend to combat and reject any preconceived notions which may nudge their way into the engineers' consciousness and which may divert the approach from the straight-and-narrow path. These calculations should confine the creative effort to the realm of practical experience and proved principles.

The calculations may require and include special analytical studies of certain aspects of the project, particularly where the record of experience is incomplete and where new basic art is involved. Engineering application of nuclear energy is a current example of new basic art wherein special, analytical studies are required as a part of creative calculations. Following completion of calculations, specifications are drawn up to govern the creative process. These specifications may be likened to the rails which keep the train confined to its path and prevent it from wandering in its approach to the desired destination.

Specifications for creative engineering should be as detailed as possible if they are to serve their useful purpose. However, they cannot foresee every possible contingency nor can they foretell the future birth of new technical thinking. Ergo, specifications should attempt to define the latitude to be allowed, the basic principles which will obtain in the making of selections and the circumstances under which the established approach may be reviewed for redirection.

Where an engineering project involves more than one technical art—and all but a very few do involve many arts—there are two or more groups of engineers making calculations and setting up specifications for their segment of the project, each independent of the others. As previously emphasized, co-ordination is an essential in the management of engineering and such co-ordination forms a part of this stage of creative engineering. The calculations and specifications of all groups are brought together in an engineering conference or by some other suitable device, conflicts are resolved, and an overall set of specifications are worked out. This usually involves some modification of calculations, compromises of technique and clarification of end objective. This integration of separate technologies may stimulate new original thinking.

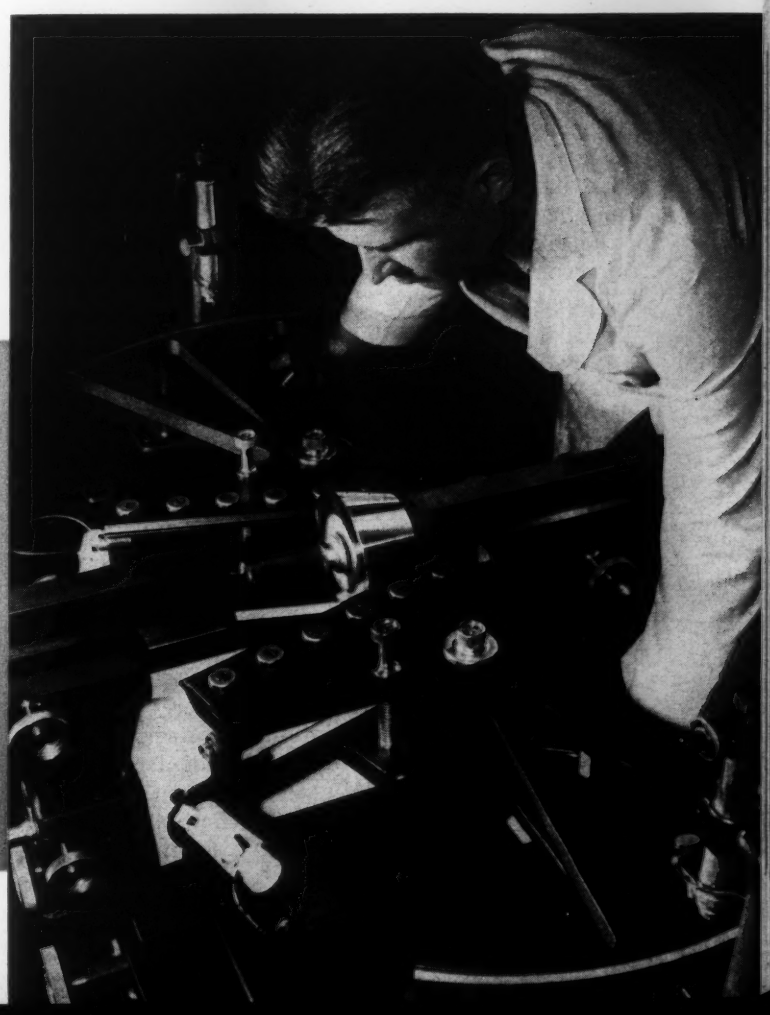
Rarely does the first set of calculations and specifications integrate nicely and much of the ground covered may have to be gone over a second, third or fourth time before all standards are met and all requirements are satisfied. This repetition may become onerous and discouraging to those who like to "get on with it" but it should be borne in mind that as many of the usual *necessary* mistakes as

Photo, courtesy Glenn L. Martin Co.

" . . . details require careful checking to insure . . . that parts will go together in the final assemblies . . . "

" . . . included also . . . is the creative engineering of special tools, jigs, fixtures, etc. . . . "

Photo, courtesy Timken Roller Bearing Co.



possible can be made in this early stage of the project with a minimum waste of engineering resources, and that the future effort is thus made more productive. It is better to make haste slowly at this point, and learn to walk before attempting to run.

Stage D—Preliminary Progress Report: The creative engineering staff is able, at this point, to report to the superior management, sponsor, client, etc., the direction which its effort will take and, whether or not such a report has been requested, it should be prepared and submitted. The sponsor should be enabled to know beyond reasonable doubt, that his interests are being well served and to interject such redirection as may be needed to assure that those interests are well served. Again, this report may generate new definition of the objective and show how the sponsor interest may be served even better than was initially expected, and result in an even better end product.

A second purpose of the progress report at this point, is to supply a record of understanding between engineering and sponsor so that there can be no future controversy regarding the approach taken or the end objective sought. Formal sponsor approval should be given at this point before further engineering effort is made.

A third and important purpose of this report is its contribution to the engineering record for the use and benefit of future and parallel engineering projects. This report becomes a part of the prior art to be studied under *Stage A* of future projects, as described in the foregoing.

Central Engineering Develops End Product

Stage E—Preliminary Design: Pencil is put to paper on the drafting board and the end product first begins to take composite form in this stage. Whereas the project has been carried on by two or more separate groups heretofore, there now appears a central engineering group which starts to put together in rough form, the structure of the end product. Composite sketches and layouts are made, correlations of the different technical arts are worked out and the framework of the end product is developed.

Although involving two or more basic technical arts, the whole project will reflect one of these arts more strongly than the others and be predominantly mechanical, structural, chemical, electrical or industrial, etc. The project engineer who heads up this central group will be selected for his greater knowledge of that predominant art as will the leaders who work under his direction. Insofar as possible, each of the basic arts involved should be well represented in the composite abilities of the group. However, the group continues to receive direction from technical specialists other than the project engineer and his assistants.

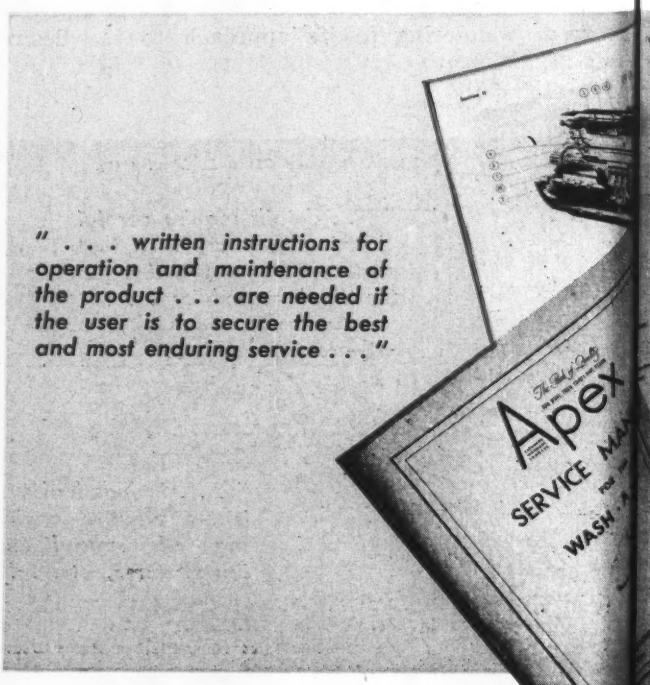
This preliminary design stage usually discloses new problems not encountered in the previous stages, particularly of physical relationship. Assume, for instance, that *Stage C* showed a gear of a certain size operating in a differential gear train but that physical interference does not show up until the scale layout drawing is made at *Stage E*. As a result, new calculations must be made and checked against the composite calculations and specifications and entered in the record. Some of the new problems disclosed in this stage may be solved only by practical experimentation as with models, "bread-board" hookups, etc. If so, sketches are made by the central creative group and these sketches are used to make up models for testing.

Project Engineer Correlates All Efforts

In addition to the central creative group, the technical specialists continue to carry out their parts of the project in collaboration with that group. The appearance design group makes its preliminary sketches in scale, color and perspective. The electrical group works out control and power circuits while the structural group refines its stress analyses, materials, shapes, fastenings, etc. The chemical group settles on temperatures, flow rates, conductors, controls, gaging and measurement devices, etc. Another group, interested in operation and performance, may develop detailed standards, performance forecasts, methods of operation and maintenance, provisions for safety, lubrication and related matters, etc. All of these specialist contributions are correlated by the project engineer with the framework evolved by the central creative group.

Stage F—Practical Experimentation: This is essentially a shop job performed by persons manually skilled in building experimental models, in testing models, in appraising test results and in interpreting test results into practical design recommendations. The process is employed where the record of practical experience is incomplete and empirical formulas are lacking or inconclusive. Hydraulic engineering frequently employs working models of streams of moving water to work out formulas for erosion, and marine engineering uses models in moving water to perfect hull design.

Again, as in each previous stage, practical experi-


" . . . written instructions for operation and maintenance of the product . . . are needed if the user is to secure the best and most enduring service . . . "
Apex
SERVICE

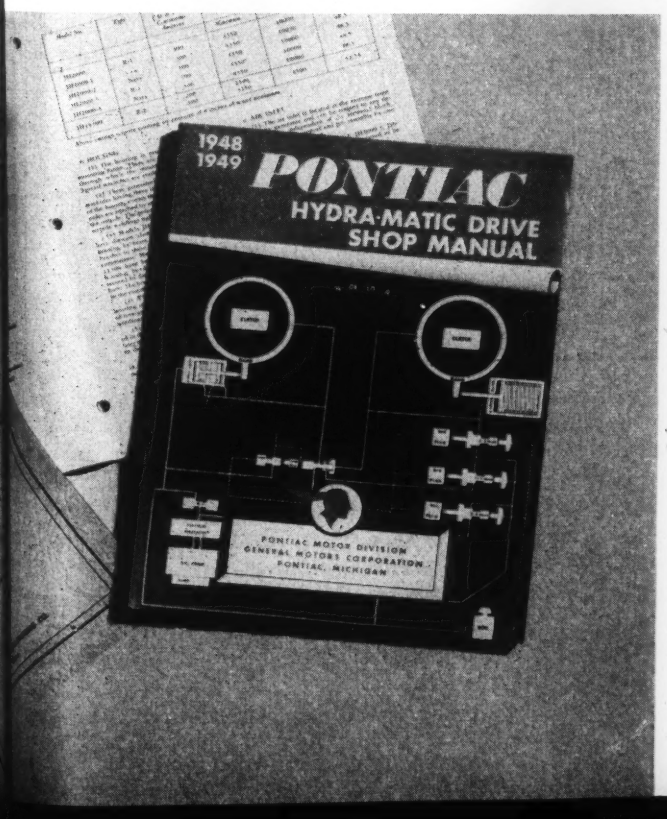
mentation may disclose basic errors in the approach and necessitate fresh calculation, alterations in specifications and preliminary designs, and a revision of record of the engineering effort. However unfortunate such errors may be, this stage is needed in many projects to persevere in the policy of committing errors as early as possible in the project and to reduce to a minimum the probability of future error in the final creative effort.

Although discussed at this particular point in the sequence of stages, practical experimentation may occur at other and more appropriate points. It should *not* be necessary at any later stage in the project.

Stage G—Intermediate Progress Report: At this point, the engineering approach is crystal clear to the creative engineering staff. They know exactly what the outcome of the project will be, how the end product will be developed, how much longer the project will take and all of the other answers. A second progress report to superior management, client or sponsor is in order at this stage.

Further alterations in the objective as defined by the sponsor are unlikely at this point, although not impossible. It is more probable that the project budget may require alteration and, if more time or funds are to be needed, the fact should now be known and can be relayed to the sponsor, together with a formal request for the supplementary budget. This request can be fully supported and justified by the exact knowledge of the project's outcome and prospect of approval of the supplementary budget should be excellent.

Again, as in the case of the previous progress report, the sponsor is formally notified of the project's outcome and gives formal approval for continuation of the project. This report lays down in all necessary detail the work to be done to complete the project and supplies the engineering staff with full instructions for completion. This report adds to the record, for future reference, the logic and evidence which supports the final engineering effort.



Stage H—Final Design: This stage contains the final and complete creative effort in which all modifications defined in previous stages are incorporated and the final form of the end product is crystallized. Layouts are completed together with appearance renderings, structural outlines, and the end products of the other creative arts involved in the project. When this stage is completed, creative engineering is also completed. It remains only to interpret and express the creative work in terms which can be executed by others.

Stage I—Design Execution: The execution of a creative engineering project consists of routine preparation of detail drawings of individual parts and assemblies, bills of materials, drawings lists, specifications of purchased parts, final appearance design renderings, instructions for special fabrication, assembly and finishing, plot and site plans, instructions to bidders and like material.

Accuracy A Primary Requirement

These details require careful checking to insure that specifications are fulfilled, that parts will go together in the final assemblies and that engineering standards are observed. There is no creative effort involved in this stage in respect to the basic technology of the end product, and this work requires no great knowledge of the basic creative arts incorporated into the end product. The prime requirement in this stage is faithful exactitude in describing the end product of engineering and the means for bringing it into existence.

Stage J—Prototypes: Some engineering projects include the building of prototypes or exact models of the end product. These may be required for several reasons such as photographs for advance sales promotion material, enabling legislation, patent applications, and many others. Again, as in *Stage F,* this is the work of shop technicians who are manually skilled in fabrication, assembly, finishing, etc., and in the photographic arts.

Stage K—Processing: The engineering project may include also the processing of the means to produce the article engineered. The processing of the product is the preplanning, in all detail, of the materials, tooling, personnel, and facilities to be employed and of the exact costs of producing the article. Processing may be required for placement of purchase orders for materials and special tools, employment of personnel of suitable caliber, purchase of machinery, land and buildings and other acts involved in reproducing the article engineered.

Also included under this heading is the creative engineering of special tools, jigs, fixtures, etc., which may be needed. Process engineering, or production engineering, is essentially creative in character and must be done in collaboration with and under the direction of the project engineer.

Stage L—Manuals of Operation and Maintenance: Where the end product of creative engineering is to be used by the layman not intimately skilled in the product's art, it is frequently necessary to supply the user with written instructions for operation and maintenance of the product and this applies to both

domestic and industrial users. Such manuals are needed if the user is to secure the best and most enduring service from the product.

Writing of manuals of operation and maintenance is another of the onerous tasks connected with creative engineering and commonly is shunned by the engineers best qualified to undertake the task. It is possible, however, for a person skilled in technical writing in general but not highly trained in the technology of the particular product to prepare manuals provided ample collaboration with the creative engineers is possible. In fact, the untrained mentality more nearly resembles the mentality of the user and is best able, therefore, to seek out and express the text of the manual in terms easily understood by the lay user. The manual of operation and maintenance denotes the distinctly lay point-of-view of the end product of engineering and is necessary to complete the whole record of the creative engineering project.

Stage M—Final Project Report: Every creative engineering project should be concluded with a final and comprehensive report not only for the information of the sponsor but for future reference in preliminary investigations for parallel projects. This final report consists of the two previous reports plus information and data gathered in *Stages J, K,* and *L,* and the text of the manual of operation and maintenance.

Plating Affects Parts Adversely

ADVANTAGES of chromium plating on gages, cylinder walls, piston rings and other machine parts, in the absence of proper precautions, may be offset by the adverse effect of the plating on the fatigue limit of the basic metal. An investigation by the National Bureau of Standards and sponsored by the Bureau of Aeronautics, Department of the Navy, provides information of interest to industries which produce or utilize chromium-plated machine parts.

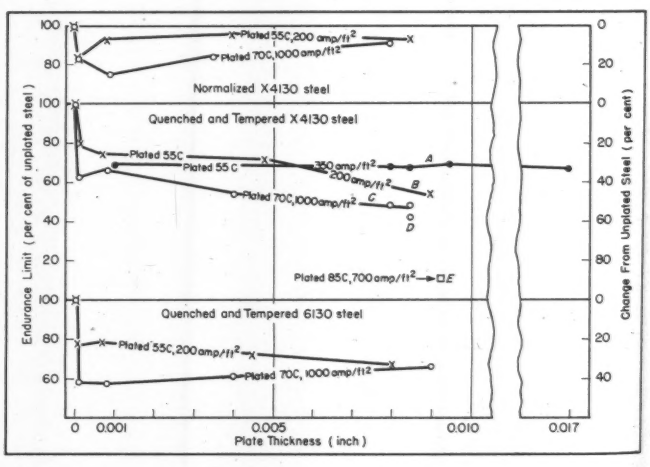
Specimens about ¼-inch in diameter were machined from three lots of SAE X4130 rod and one lot of SAE 6130 steel rod. The X4130 steel was either normalized to a Rockwell hardness of 90-B or quenched and tempered to a hardness of about Rockwell 40-C, while the 6130 steel was quenched and tempered to a Rockwell hardness of 33-C. After grinding, polishing, and chromium-plating, the specimens were subjected to fatigue tests in the R. R. Moore type rotating-beam machine operating at 1800 or 3600 rpm. Eight or 10 specimens usually were required to obtain the fatigue limit for any one set of conditions.

In all cases chromium plating was found to reduce the fatigue limits of the steels studied, although the effect was less pronounced under some conditions than others. In general, the reduction in fatigue limit increased with increased hardness of the steels. For steel of a given hardness, the fatigue limit decreased with increased temperature of the plating bath. While plate thickness appeared to have little effect on the fatigue limit of specimens plated in a bath at 55 C and a current density of 350 amp per square foot, no generalizations could be made regarding the effect of this factor at other current densities and temperatures.

It was discovered that the fatigue limits of specimens plated and subsequently ground to remove a part of the plating were equal to or greater than those of specimens initially plated to the same thickness as the ground specimens and tested as plated. Other experiments showed that interruptions of the plating process did not reduce the fatigue limit of the plated specimens provided proper precautions were taken for continuing the plating.

Chromium-plated objects often are heated to temperatures between 90 and 200 C with the object of improving their mechanical properties by expelling the hydrogen deposited with the chromium. A systematic study was therefore made of the effect of heating on the fatigue limit of chromium-plated steel. The results showed that the fatigue limits of quenched and tempered specimens heated after plating decreased to a minimum value for some heating temperature between 100 and 300 C and thereafter increased with increased heating temperatures. Fatigue limits of specimens heated for 1 hour at 440 C in some cases were 87.5 per cent of that of the unplated steel, whereas fatigue limits of specimens plated but not heated, and of specimens heated 1 hour (in air) at 193 C, were only about 68 and 27 per cent, respectively, of those of the unplated steel.

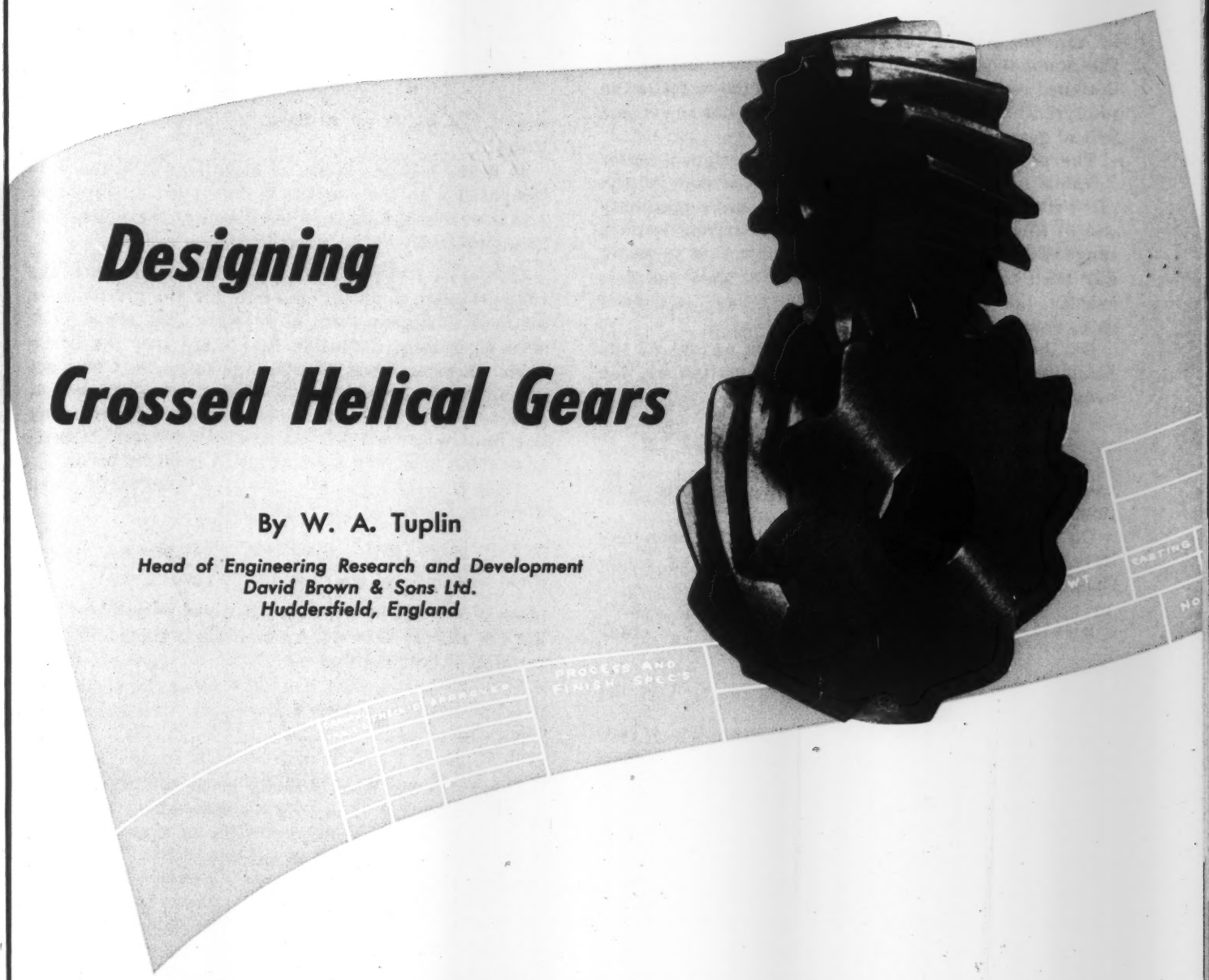
Three possible causes for the adverse effect of chromium plating on the fatigue limits of the steels were (1) the embrittling effect of hydrogen deposited with the chromium, (2) cracks in the chromium, and (3) residual stresses in the chromium. However, it was shown that there was no simple relationship between the fatigue limit and the amount of hydrogen



remaining in the chromium after a baking treatment. Further experimental evidence indicated that cracks present in the chromium were not the principal cause of the reduced fatigue limits of the plated steels.

It is generally agreed that residual stresses in a material markedly affect its fatigue limit. Thus, in steel, compressive stresses, such as are produced by shot peening, increase the fatigue limit; tensile stresses have the opposite effect.

Designing Crossed Helical Gears



By W. A. Tuplin

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Need for time-consuming, exact solutions to crossed helical gear problems can be avoided by making use of the flexibility of the involute tooth form system

ALTHOUGH gearing text books rightly emphasize the flexibility of the involute system, none makes use of that flexibility to avoid the need for solving exactly the crossed helical or so-called spiral gear problem. This problem is to determine the helix angles of a pair of cylindrical gears, *Fig.* 1, connecting nonparallel, nonintersecting shafts set at a given angle and center distance.

The essential condition—that the sum of the pitch radii of generation of the two gears shall be equal to the working center distance—may be expressed in the form

$$\frac{np_n}{\pi}\sec\psi + \frac{Np_n}{\pi}\sec(\Sigma-\psi) = 2C \quad \ldots\ldots\ldots\ldots (1)$$

where symbols are as given in the Nomenclature and notation is as in *Fig.* 1. In Equation 1, pitch diameter of generation of the pinion is

$$\frac{np_n}{\pi}\sec\psi$$

and pitch diameter of generation of the gear is

$$\frac{Np_n}{\pi}\sec(\Sigma-\psi)$$

Equation 1 looks as if it should be soluble by a direct trigonometric process, but actually it is not. Much ingenuity has been expended by many writers on turning what is essentially a trial and error process into different forms.

The condition expressed in Equation 1 is an unnecessary restriction if the gears are to be of involute form, produced by generating cutters. It is well known that the designed center distance of a pair of

spur or helical gears need not be equal to the sum of the pitch radii of generation. In other words, the nominal pitch radii (whose sum *is* equal to the designed center distance) need not be the same as the pitch radii of generation. The same applies to crossed helical gears.

The permissible range of ratio of designed center distance of crossed helical gears to the sum of the pitch radii of generation is in many cases remarkably wide. For the present purpose a relatively narrow range will suffice since the only object is to avoid any need to solve the crossed helical gear problem exactly. In this article a "useful range" is defined in a convenient way.

On the basis of design that makes no use of the flexibility of the involute system (Equation 1), the nominal pitch diameter of the gear is

$$D = 2C - \frac{np_n}{\pi} \sec \psi = \left[\frac{p_n}{\pi} \sec (\Sigma - \psi) \right] N$$

and a value of ψ must be found to satisfy this equation.

On the basis used here, D in the design stages may have any value between the results given by Equations 2 and 3:

$$D = \frac{p_n}{\pi} \sec (\Sigma - \psi) \left[N - 1 + \frac{17 \cos^3 (\Sigma - \psi)}{N - 1} \right] \quad \ldots\ldots (2)$$

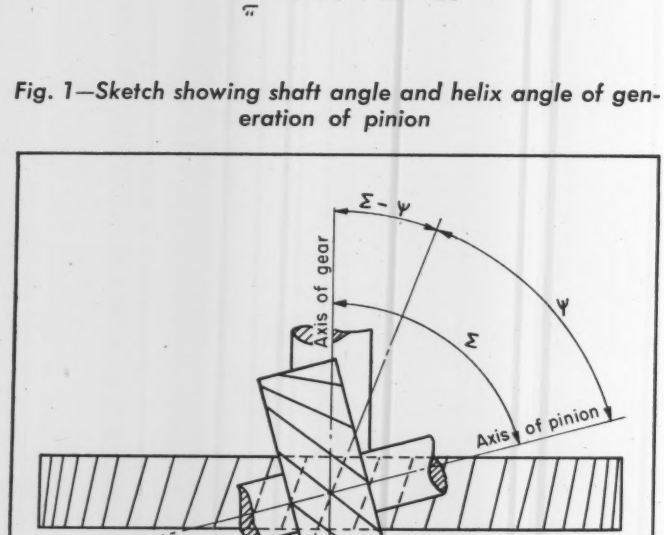
$$D = \frac{p_n}{\pi} \sec (\Sigma - \psi) \left[N + 2 - \frac{17 \cos^3 (\Sigma - \psi)}{N + 2} \right] \quad \ldots\ldots (3)$$

These limits define a range of D within which generation by a 20-degree full-depth cutter will produce satisfactory tooth shapes. The manner of defining the range is arbitrary; the range itself is not, in general the widest possible but it does meet the present purpose.

Corresponding to Equation 1, the essential condition is now

$$\frac{np_n}{\pi} \sec \psi + D = 2C$$

Fig. 1—Sketch showing shaft angle and helix angle of generation of pinion



In other words, nominal pitch diameter of the gear

$$D = 2C - \frac{np_n}{\pi} \sec \psi \quad \ldots\ldots (4)$$

where the value of D must lie between Equations 2 and 3.

It is the latitude given by Equations 2, 3 and 4 as compared with the exactitude demanded by Equation 1 that evades the rigor of the classical crossed helical gear problem.

GUIDANCE RULES: With crossed helical gears, the range of pitch diameters possible for any given center distance and gear ratio is so wide that some guidance in making tentative first steps may be desirable. In some cases, external considerations may set an upper or lower limit to the pitch diameter of pinion or gear. In such cases a value approaching the limit may be taken as a starting point. Otherwise some arbitrary starting point must be used.

Since $D = (Np_n/\pi) \sec (\Sigma - \psi)$, approximately, and $d = (np_n/\pi) \sec \psi$, it is found that

$$\frac{D}{d} = \frac{N}{n} \frac{\cos \psi}{\cos (\Sigma - \psi)} = R \frac{\cos \psi}{\cos (\Sigma - \psi)}$$

since N/n must be equal to the given ratio R. Writing $\cos (\Sigma - \psi) = \cos \Sigma \cos \psi + \sin \Sigma \sin \psi$ and rearranging, it is found that

$$\tan \psi = \frac{\left(R \frac{d}{D} - \cos \Sigma \right)}{\sin \Sigma} \quad \ldots\ldots (5)$$

If a value can be arbitrarily given to d/D, the corresponding value of ψ may be determined by Equation 5. Such an arbitrary value is $d/D = \sqrt{1/R}$, which causes Equation 5 to become

$$\tan \psi = \frac{(\sqrt{R} - \cos \Sigma)}{\sin \Sigma} \quad \ldots\ldots (6)$$

Also

$$d = \frac{2C}{1 + \sqrt{R}} \quad \ldots\ldots (7)$$

and since $d = (np_n/\pi) \sec \psi$, then

$$np_n = \pi d \cos \psi = \frac{2\pi C}{1 + \sqrt{R}} \cos \psi \quad \ldots\ldots (8)$$

The aim must now be to discover a combination of suitable values of n and p_n that satisfies Equation 8, remembering that (1) ψ may have any value approximating that derived from Equation 6, (2) p_n must be a standard pitch, and (3) there must be a whole number N such that N/n is within the tolerance on R.

It is then ascertained whether $2C - np_n/\pi \sec \psi$ lies between the values of Equations 2 and 3. If not, other values of ψ must be tried until one is found that satisfies the condition.

BLANK DIAMETERS: The addenda are determined from the "equivalent numbers of teeth", $n \sec^3 \psi$ and $N \sec^3 (\Sigma - \psi)$. Let the gear with the smaller equivalent number of teeth be called No. 1 and the other No. 2. Let the ratio of the smaller equivalent number to the larger equivalent number be u. Then, satisfactory

Nomenclature

n, N = Number of teeth on pinion and gear, respectively
p_n = Normal pitch of generating cutter used for pinion and gear
ϕ_n = Normal pressure angle of generating cutter
C = Designed center distance
Σ = Shaft angle
ψ = Helix angle of generation of pinion
d = Nominal pitch diameter of pinion. Equal to pitch diameter of generation of the pinion
D = Nominal pitch diameter of gear. Equal to $2C - d$ but *not*, in general, equal to the pitch diameter of generation of the gear.
e = ½ Deviation of $2C$ from Equation 1 value
a, A = Addenda of pinion and gear, respectively
b, B = Dedenda of pinion and gear, respectively
d_0 = Tip diameter of pinion = $d + 2a$
D_0 = Tip diameter of gear = $D + 2A$
f, F = Face width of pinion and gear, respectively
R = Gear ratio = N/n
u = Ratio of smaller equivalent number of teeth to larger equivalent number

formulas for addenda are:

Addendum for No. 1 = $(p_n/\pi)(1.4 - 0.4u)$
Addendum for No. 2 = $(p_n/\pi)(0.6 + 0.4u)$

The blank diameter for each gear is determined by adding twice its addendum to its nominal pitch diameter.

If the cutter is sunk to standard full depth into blanks of these diameters, the resulting gears would mesh together at center distance C with no backlash, if everything were perfect. As that never is the case, and as the condition of no backlash is one to be avoided wherever possible, the cutter should be sunk into each blank to standard full depth plus an amount corresponding to the desired backlash.

In crossed helical gears the backlash, defined as free circumferential movement at the pitch cylinder of the teeth of one gear as limited by those of the other gear, is generally different for pinion and gear. It is preferable to define backlash as the difference between tooth thickness of one gear and space width of the other, measured at the nominal pitch cylinder in the normal section. A reasonable amount for such backlash in crossed helical gears is what would be allowed in spur gears of the same normal pitch.

The excess of cutting depth over standard depth, for both gears, is 0.25 cot $\phi_n \times$ desired normal backlash, whether or not $e = 0$. The value of e as defined in the Nomenclature and from Equation 1 is:

$$e = C - \frac{p_n}{2\pi}[n \sec \psi + N \sec (\Sigma - \psi)]$$

In this respect crossed helical gears differ from spur and parallel-shaft helical gears; in gears of these latter types there is backlash between teeth that have been cut to full depth in blanks made to suit a center distance for which e is not zero.

LEAD: Whatever operating center distance is used, the leads of the tooth spirals of the two gears are

np_n cosec ψ and Np_n cosec $(\Sigma - \psi)$

FACEWIDTH: In order that contact between the teeth of crossed helical gears shall be limited by the tips of the teeth and not by the end faces, the width of each gear should be at least

$$2.5 \cot \phi_n \left[g \sin \psi_c + \frac{|e| \sec \psi_c}{\tan \psi + \tan (\Sigma - \psi)} \right] \quad \ldots\ldots (9)$$

where g = the greater of the addenda of gear and pinion, ψ_c = helix angle of the gear under consideration, and $|e|$ = numerical value of $C - (p_n/2\pi) \times [n \sec \psi + N \sec (\Sigma - \psi)]$ which is always taken as positive. Very often, e is small enough to neglect.

NUMERICAL EXAMPLE: The following calculations have been made by using an ordinary 10-inch slide rule; its accuracy is ample for the purpose.

Let it be required to determine the dimensions of crossed helical gears to work at a center distance of 4 inches ±0.002-inch, *Fig.* 2, with shaft angle 75° ±1′ and to have a ratio of 3.7 ±0.05. The tolerances on center distance and shaft angle are regarded as manufacturing tolerances and not used in design. The tolerance on ratio may be used in design as none is required in manufacture, specified numbers of teeth being exactly attainable.

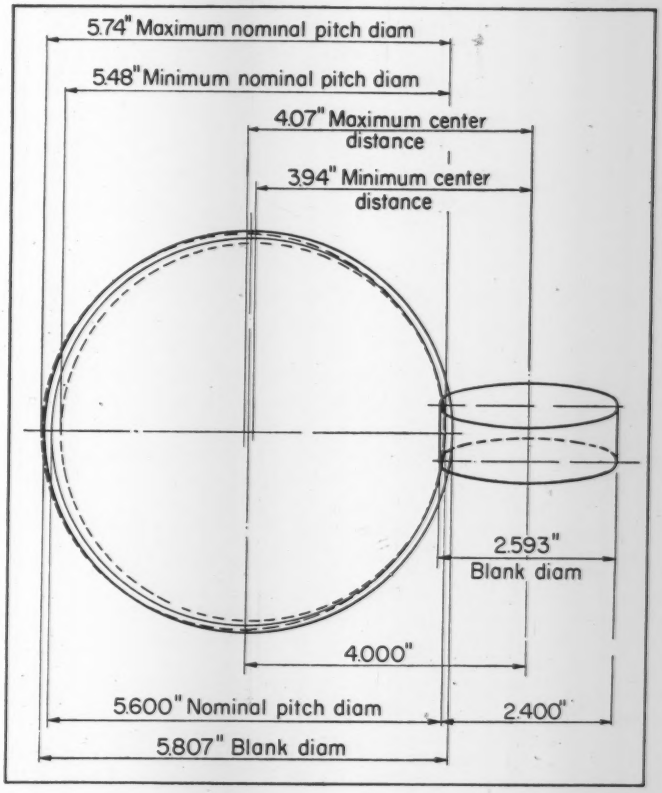
With slide-rule set at 3.7, a few ratios that are seen to be within the tolerance are 48/13, 52/14, 59/16, 63/17, 70/19 and 74/20. From Equation 6

$$\tan \psi = \frac{(\sqrt{3.7} - \cos 75°)}{\sin 75°}$$

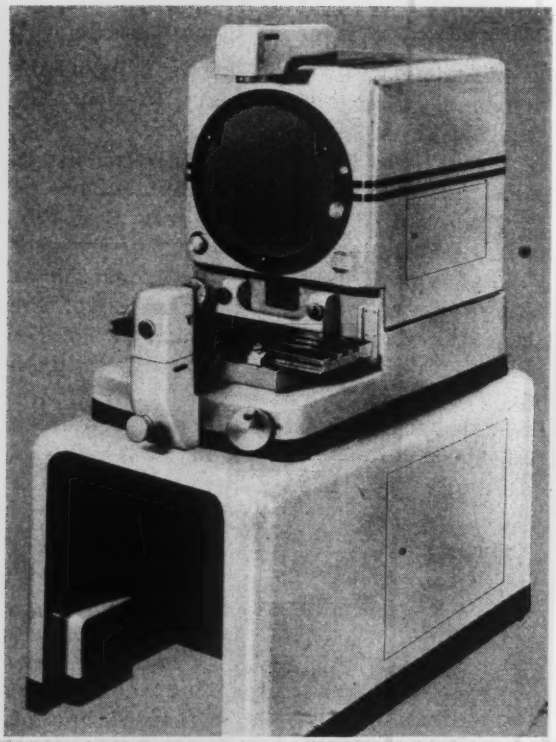
$$= \frac{(1.923 - 0.258)}{0.966} = 1.725$$

(*Continued on Page* 202)

Fig. 2—Drawing of crossed helical gears with dimensional notation developed in the numerical example



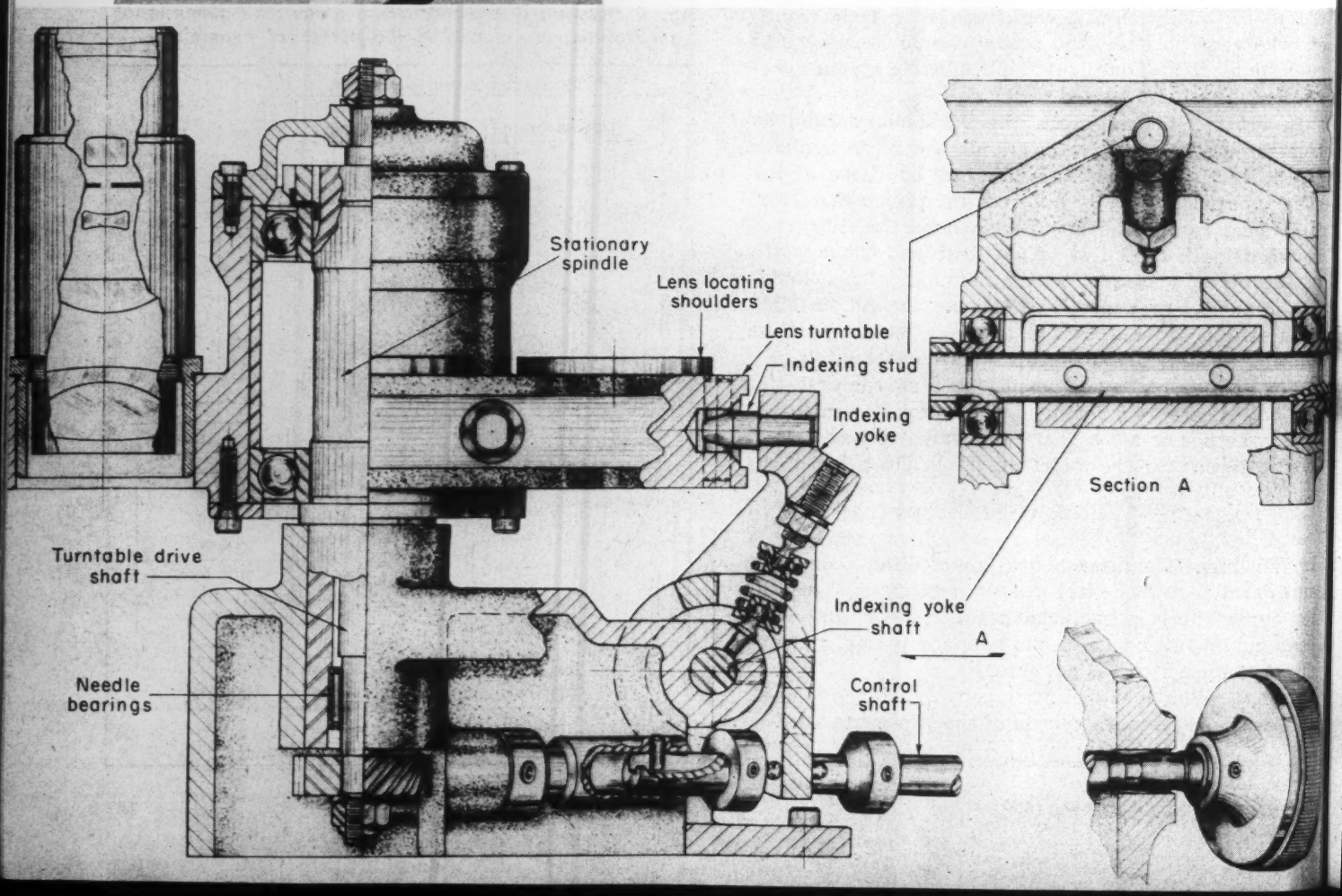
Contemporary DESIGN



Six-Lens Contour Projector

ALIGNMENT between lens barrel and turntable of the Eastman Kodak Co. contour projector, Model 2, shown at the left, requires precision tolerances and a highly accurate lens turntable locking mechanism. In designing the lens turret, it was necessary to provide a rigid turntable which would position each of the six lenses accurately on the instrument's optical axis. Parallelism between the axis of any lens and the optical system is held to within 1 minute of angle to provide uniform magnification over the whole screen area. Relative position of the lens, with regard to the object and screen, is held to within 0.002-inch to provide correct focus and magnification at rated power.

Base of the turret supports a hollow stationary spindle



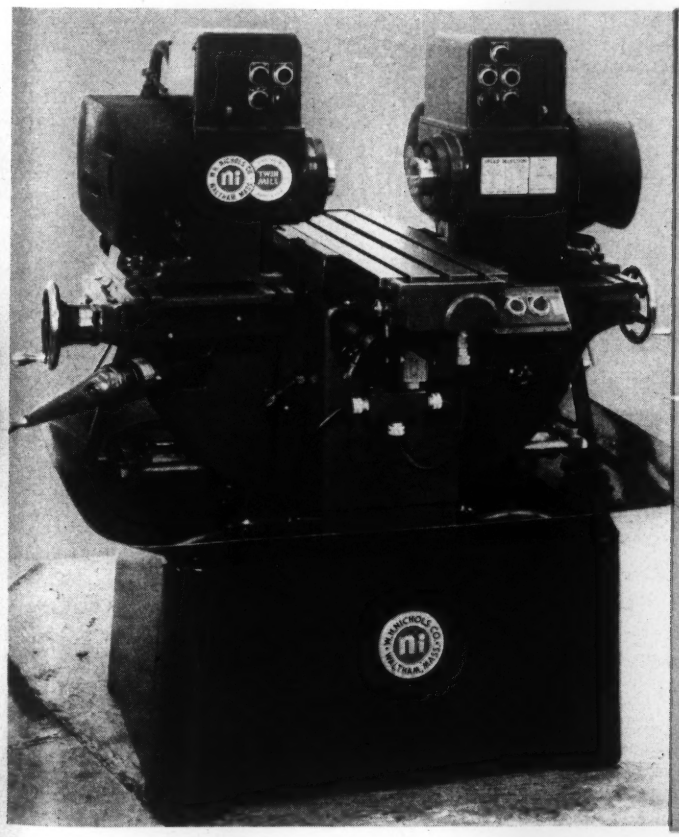
which carries the lens turntable on preloaded ball bearings, drawing bottom of previous page. Position of the turntable in the vertical plane is rigidly fixed by resting the inner race of the lower bearing on a collar on the spindle and by eliminating all backlash through preloading. Lenses are positively seated on brass locating shoulders held to within 0.0003-inch per inch face squareness. Needle bearings absorb any radial thrust imparted to the vertical turntable shaft. This shaft is connected through helical gears to the operator's hand knob, which is used both to rotate the turret and to actuate the detent.

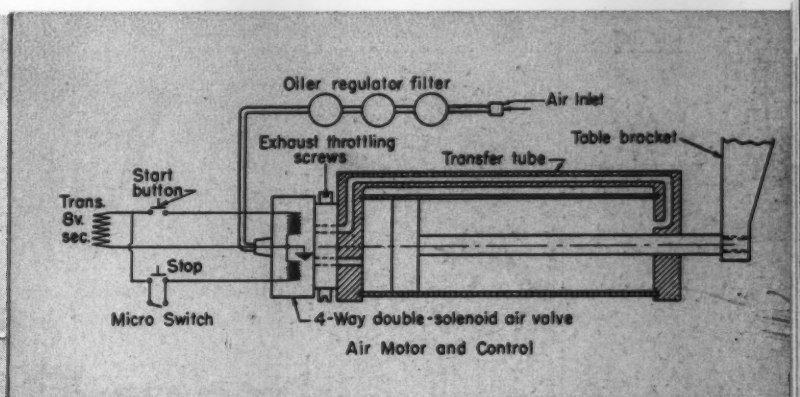
This detent mechanism consists of an indexing stud carried on a yoke which rotates on a shaft on preloaded bearings. When the hand knob is pushed in, a collar on the hand knob shaft acts against a button on the yoke to disengage the stud and permit the turntable to spin free. With the knob pulled out, another collar acts against the yoke to engage the stud in its seat on the turntable. The stud end is tapered and seats itself between two hardened pins under pressure of a spring. Locating accuracy of the turntable, vertically and angularly, is such that the projector magnification can be changed to any one of the six values provided (10× to 100×) without refocusing.

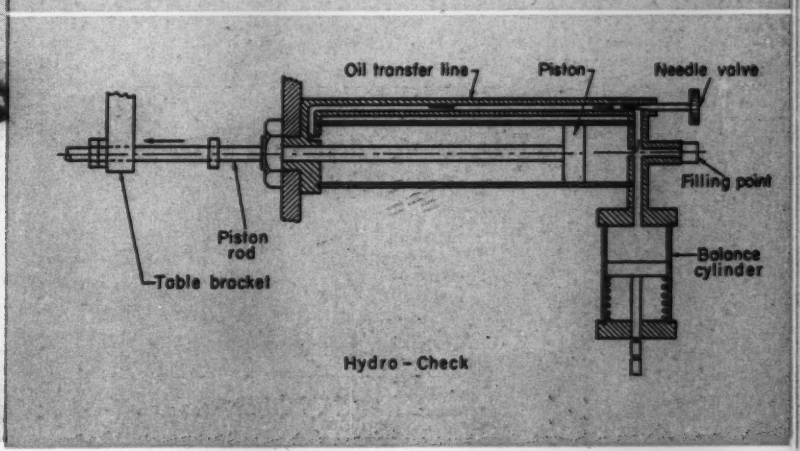
Versatile Dual-Head Miller

UNUSUAL flexibility in the milling of small parts is provided in the design of the dual-head milling machine shown below. Both heads of the Twin Mill, made by W. H. Nichols Co., are adjustable parallel to the direction of table travel, as well as vertically and longitudinally. Powered by an individual motor for each unit, the heads include oil-lubricated gear boxes providing 15 speeds from 55 to 2050 rpm. Keys sliding in T-slots in the saddle make an 8½-inch adjustment possible in the direction of table travel. Conventional knee and saddle slide movements are also obtainable.

The machine table is arranged for automatic operation, powered by an air-hydraulic system providing rapid approach to cutting position and infinitely-variable hydraulically controlled cutting feed. The automatic table drive is pneumatic, solenoid-controlled, with trips and limit switches to govern table movement. Pressing the start button energizes one of the solenoids controlling a four-way air valve,







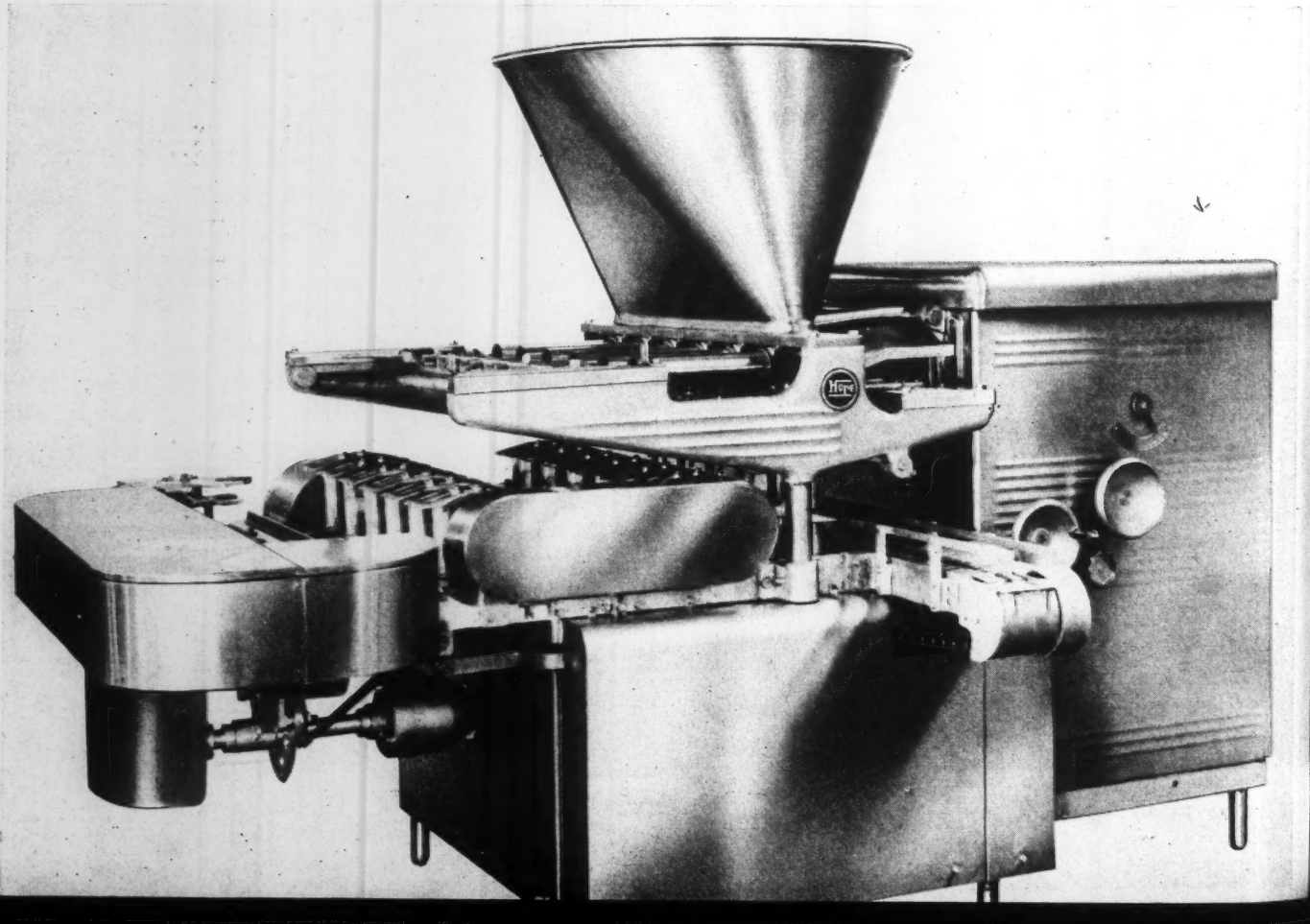
upper drawing previous page, which admits air to one end of the cylinder, driving the table forward. Length of stroke is determined by the setting of a dog on the table which trips a Micro Switch, energizing the other solenoid and reversing cylinder travel.

Speed of the cutting is controlled and table chatter is eliminated by a Hydro-check cylinder unit, lower drawing, previous page, in which the piston rod and piston are actuated by movement of the table. As the table advances, oil is forced ahead of the piston and out through the transfer tube at a rate controlled by a needle valve. Rate of oil flow through this needle valve controls piston speed and therefore, speed of table travel. Oil passes through a one-way valve in the piston on the return stroke of the table. Further versatility is provided by right-angle milling attachments which can convert both heads from horizontal to vertical milling.

Container Cleaning and Filling Machine

ISOLATION of the power unit from the filling side of the container filling machine, below, permits easier maintenance of both sections and results in a more sanitary filling operation. The Hope Machine Co. Sanitype 20 machine includes, from left to right, an infeed conveyor, a double turnover and cleaning device, the filler, an outfeed conveyor, and the power section. Sequence of operation is as follows: Containers moving in on the conveyor are positioned into six filling lines and turned over 180 degrees by the first rotating pocket. The upside-down containers are cleaned by air blast and vacuum heads mounted under the table surface. Clean containers are then turned upright by a second rotating pocket. The jars are next placed on a rising table positioned under the filling nozzles.

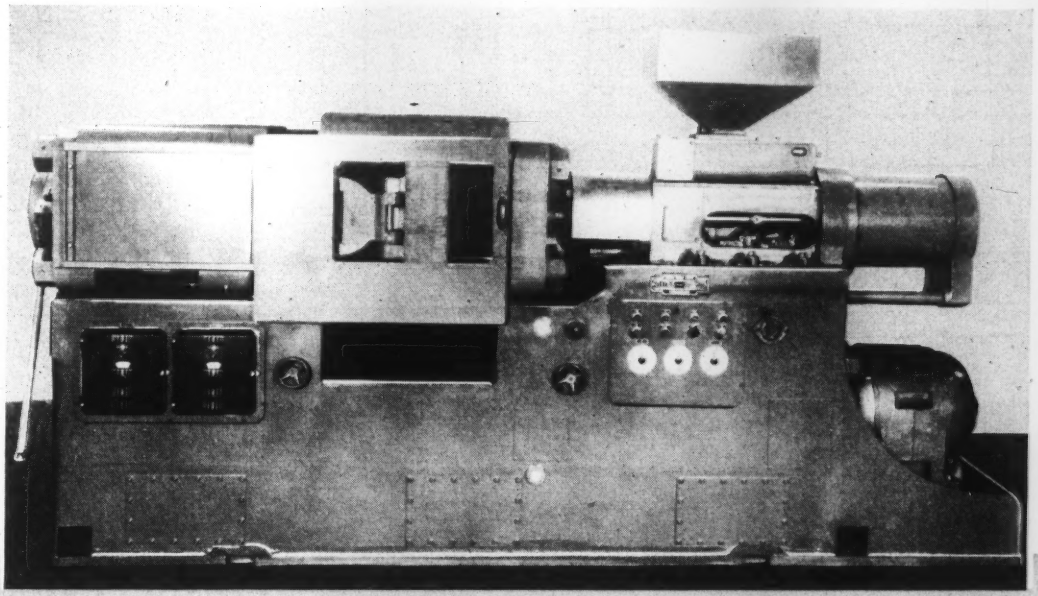
If a jar is missing in any of the six filling stations, an automatic control prevents a discharge at that position. The discharge conveyor delivers the filled containers to the side ready to be capped. Features incorporated into the design include an adjustable filling height, variable rising table movement and a speed range of 72 to 120 containers per minute (on the six-line model).

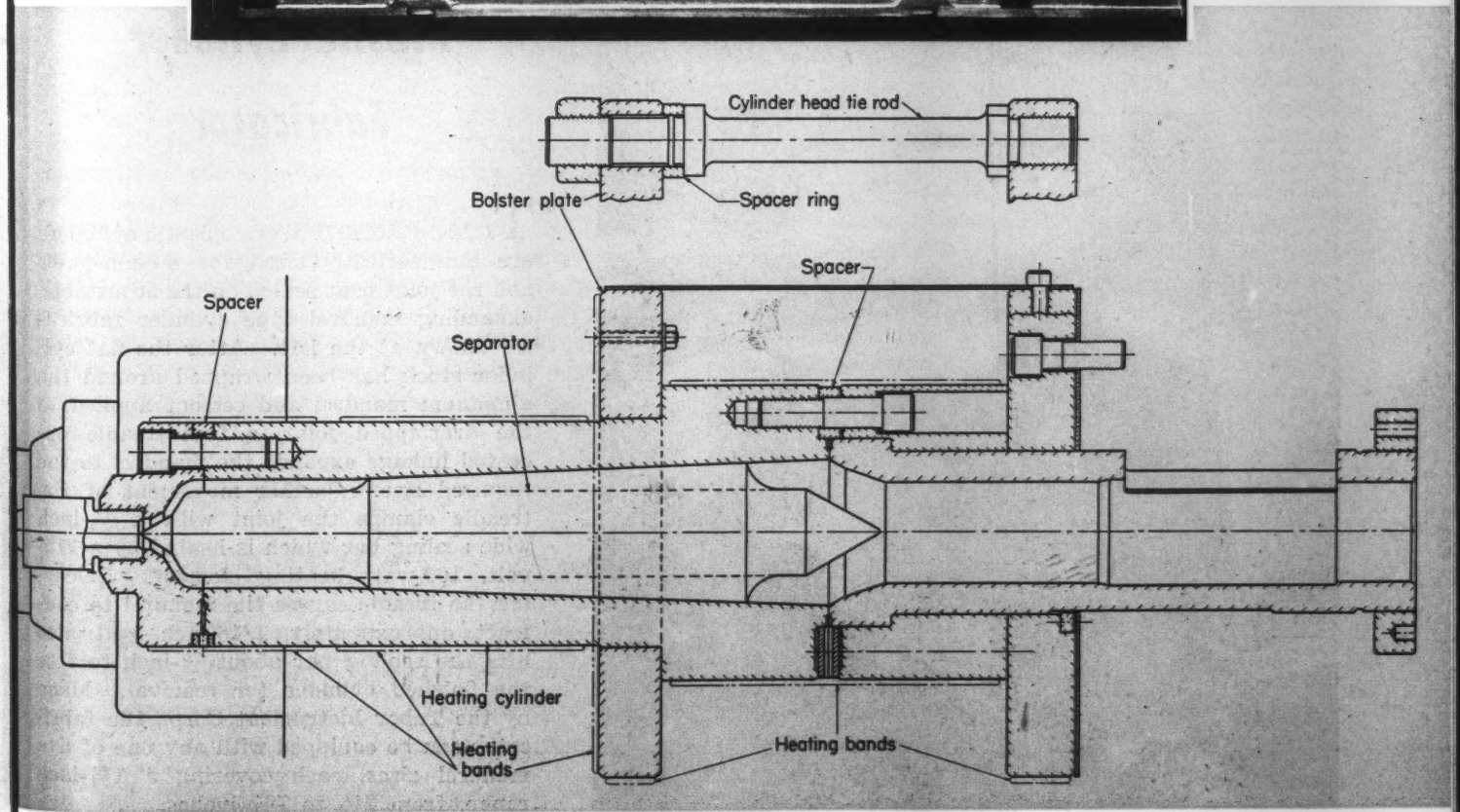


Injection Molder With Novel Heating Cylinder

TWO major design problems in plastics injection molding machinery are overcome in The Fellows Gear Shaper Co. Model 5C-8 machine shown below. The first of these, breaking the plastic into small cross sections or streams for better heat transfer, is solved by inserting a tapered separator in the heating cylinder. A series of flutes on the separator provides the necessary breakup of the plastic. The tapered construction results in better heat transfer to the plastic because of the more intimate contact between separator and cylinder wall.

Dwelling or hang-up of the material in the heating cylinder, the second major problem, is prevented by the close contact between separator and cylinder wall, and by joints which will resist pressures of 20,000 psi at 400 to 600 F temperatures. Joint design, as shown on the drawing below, includes spacers which give a bridge type clamp on the lapped sealing surfaces. Electric heating bands the length of the heating cylinder are thermocouple-controlled. This machine is capable of operating at 6 cycles per minute and up to 80 lb per hour of polystyrene parts.



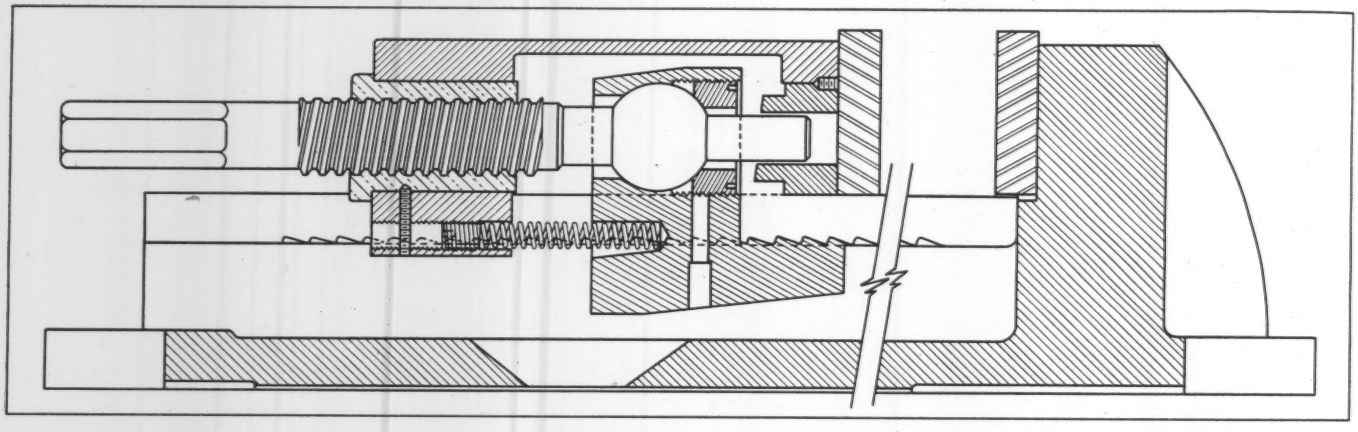


Contemporary DESIGN

Quick-Locking Vise Mechanism

QUICK operating vise shown in the drawing, below, uses a tilting block arrangement to permit free sliding of the movable jaw toward or away from the workpiece. The tilting block floats on a spherical section of the operating screw, with a spring tending to engage rack teeth on the block with teeth in the vise rails. Backing off on the operating screw disengages the teeth and allows the movable jaw to slide freely; tightening the screw tilts the block so that teeth on block and rails engage to take the tightening load.

Ames Precision Machine Works makes the vise with jaw width of 8 inches, jaw depth of $2\frac{7}{16}$ inches and maximum opening of 11 inches.



Plastic Cylinder Fabricator

TRANSPARENT thermoplastic cylinders are automatically sized for diameter and the joint heat sealed on the adjustable, expanding mandrel type cylinder fabricator shown at the left. After the flat cylinder stock has been wrapped around the aluminum mandrel and cement applied to the overlapped joint, a foot treadle operated linkage expands the mandrel to the required size. Further movement of the treadle clamps the joint with a ½-inch wide sealing bar which is heated by a 115-volt, 100-watt heating element. Releasing the treadle causes the mandrel to contract approximately 1/16-inch and also lifts the sealing bar about ⅞-inch to free the finished cylinder for removal. Made by the Tabor Instrument Corp., the fabricator can be equipped with any one of five mandrel sizes, each covering a 1¼-inch range, from 2½ to 7¾ inches.

BOLTS

HOW TO PREVENT THEIR LOOSENING

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ALTHOUGH loosening of threaded fastenings in mechanical assemblies subject to vibratory loads is a generally accepted possibility, little reproducible experimental information is available to the designer. This is rather surprising not only because of the widespread use and basic importance of threaded connections but also because many solutions to the problem of loosening in the form of locknuts or other types of locking devices have been advocated. Intelligent use of such devices frequently assists in overcoming the loosening problem, but at the same time introduces certain disadvantages such as increased assembly time, weight and cost.

Some data on the vibration loosening process of ordinary threaded connections have recently been presented by Goodier and Sweeney[1]. Together with this data, obtained by slow repeated use of a standard tensile testing machine, they have advanced an interesting theory of the loosening process based in essence on the radial "panting" action of a nut-bolt combination when subject to dynamic loading. It would appear well worthwhile to re-examine this theory in light of experimental results on the loosening phenomena that were obtained not under quasi-static conditions but under vibratory conditions more representative of actual practice. The first and necessary task, therefore, is to provide such data.

This article provides specific information on the loosening effect of such variables as number of load cycles, initial bolt tension or tightening torque, dynamic-static load ratio, misalignment, surface smoothness and load relaxation.

To subject threaded connections to a vibratory load, a Sonntag universal fatigue testing machine (1000 lb capacity) was used. The construction and operation of this machine is described in detail by Lazan[2], and its application to vibration loosening of threaded connections has been reported by Sauer and Roos[3]. The machine readily lends itself to this type of study by reason of its ability to superimpose any desired constant-amplitude sinusoidal-fluctuating load upon any desired static preload.

Adaptation of the tension-compression fixture of the Sonntag machine to accommodate the nut-and-bolt asembly is illustrated in *Fig.* 1. Both steady and fluctuating loads are applied to the assembly through the reciprocating platen to which the bolt is attached. The reciprocating platen forms the top of a cage within which an eccentric mass rotates at constant angular velocity of 1800 rpm. The horizontal component of the centrifugal force produced by the rotating eccentric is absorbed by flex plates which permit only vertical motion of the cage, and the useful vertical component is then a sinusoidal-alternating load, the maximum value of which may be varied by adjusting the eccentricity of the calibrated mass. To the bottom of the cage are fastened two preload and

[1] Numbers refer to bibliography at end of article.

TO DETERMINE the effect of alternating cycles of loading, dynamic-static load ratio, contact-surface condition, misalignment, tightening torque, and load relaxation on loosening of threaded connections, the reported tests were conducted by the authors. Recommendations for preventing loosening include specification of dynamic-static load ratios of approximately 0.7 or less, control of contact surfaces to have high frictional resistance and assembly to avoid misalignment

Fig. 1—Tension-compression fixture on fatigue testing machine adapted for loosening tests

inertia-compensating springs. The extension of these springs is a measure of the initial static tension.

Also shown in *Fig.* 1 is the optical lever system used to measure the loosening of the nut relative to the bolt. A beam of light focused from one point light source is directed to a small mirror clamped on the nut specimen and reflected to a curved scale. The angle of turning of the nut is then half the angle indicated by the movement of the light point on the scale. In some experiments, a second point light source was used to direct a beam to a mirror, attached to the end of the bolt. From the readings of the position of the reflected images, the relative rotation between nut and bolt is obtained.

With few exceptions the fastenings tested were zinc and cadmium-plated steel nuts, 5/16-inch diameter, Standard National Coarse 18 threads per inch and 4-inch long unhardened carbon-steel cadmium-plated bolts. The exceptions were one group of rolled-thread heat-treated bolts tested with regular cadmium-plated nuts. All specimens conformed to Class 3 fit gages.

Various series and types of tests were run to determine specifically the effect of certain variables. A brief tabular summary of the various tests showing the controlled dynamic and static-load values, the dynamic-static load ratio, the number of specimens tested, the condition of mating thread surfaces of nut and bolt, and the type of plating and seating surfaces is given in TABLE 1. In the early part of each test, readings of the nut rotation were usually taken every several hundred cycles; thereafter, at greater intervals. The tests were usually discontinued when additional loosening became difficult to detect. This occurred for most specimens prior to 20,000 cycles, but in some instances, observable loosening still occurred after 50,000 cycles.

NUMBER OF ALTERNATING LOAD CYCLES: In the first series of tests, (Series *A*) the specimens (5/16-18, cadmium plated) were statically loaded to 500 lb corresponding to a bolt tensile stress of 9500 psi based on a cross-sectional area calculated from the mean root-pitch diameter. In Test No. 2, a dynamic load of 400 lb was superimposed upon the static load, and in Test No. 3 a dynamic load of 450 lb was used. These loadings correspond to a dynamic-static load ratio of 450/500 = 0.90 and 400/500 = 0.80. In the 450/500 test, the bolt stress varied from a minimum of 1000 psi to a maximum of 18,000 psi through one complete cycle.

In Test No. 3 (the 450/500 tests), data were obtained for 17 different specimens, each succeeding test being run on a fresh nut and a fresh bolt of supposedly identical types. The plotted data in the form of curves for loosening vs. cycles of dynamic loading are shown for 10 separate specimens in *Fig.* 2. Two facts are immediately evident: first, the rate of loosening is initially large but usually falls off rapidly in the region from 2000-3000 cycles and, secondly, there is considerable scatter in the results.

In Test No. 2 (the 400/500 tests), twenty separate nut-bolt combinations were run. The data are presented for 10 different specimens in the form of loosening curves in *Fig.* 3. These curves show a somewhat similar decline in the rate of loosening in the 2000-3000 cycle range and also indicate that the specific amount of loosening varies greatly with the individual specimens. Complete data for all specimens in *Figs.* 2 and 3 are given by Lynn[4] and Lemmon[5].

Variations Are Not Caused by Bolt Rotation

The question arises as to the cause of the wide scatter and whether it could be due to variable amounts of bolt rotation. In the foregoing series of tests, bolt movement as such was not measured on the assumption that bolt rotation would not occur since the bolt head of each specimen was clamped before each run was started. However, to make certain that the scatter in the results was not caused by variable bolt rotation, five additional specimens—for each of the dynamic-static load ranges previously investigated—were tested with mirrors attached to the end of the bolt, and second optical lever system was used to record its rotary motion. No bolt rotation was observed, yet the inconsistencies in the amounts of loosening persisted. This is illustrated by *Fig.* 4, which shows the family of loosening curves obtained for the nut-bolt specimens of Test No. 3*a*.

DYNAMIC-STATIC LOAD RATIO: In the two groups of tests described, the dynamic-static load ratio was varied but the effect of this variation was masked by the wide scatter in the test results. To ascertain, therefore, whether variable dynamic-static load ratio did produce an effect, it was necessary to average the two aforementioned families of loosening curves. These average curves based on data for all specimens tested are shown in *Fig.* 5 and do indicate a greater loosening to accompany the larger dynamic-static load ratio. One may question whether this condition is representative of the entire load range as only two dynamic-static load ratios were used. However, additional experiments to be described later show that the trend as indicated is general. It is difficult to make any positive statement about the average rate (rather than the amount) of loosening except to note that for both the 0.9 and 0.8 dynamic-static load ratios, the rate of loosening is initially large and then decreases rapidly usually to some small value.

CONDITION OF CONTACT SURFACES: The effect of surface condition was first investigated by re-running both the 400/500 and the 450/500 tests with the individual specimen assembly consisting of a previously once-used nut mounted on a bolt in such a manner that its threads were in contact with bolt threads not previously used. "Once-used" here means that the nut was previously subject (for some 20,000 to 48,000 cycles) to an alternating sinusoidal load of 400 or 450 lb (alternating stress of 7600 or 8500 psi) superimposed on a static load of 500 lb (static stress of 9500 psi).

Loosening tests (No. 2*b*) were run in this manner for nine of the 400/500 series specimens and for eleven of the 450/500 specimens (Test 3*b*). Specific data as to which of the previously used specimens were selected as well as the actual loosening data, are given in full elsewhere[4, 5].

It is sufficient for our purposes to show only the "average" loosening curves (*Fig.* 5) for each of the two different dynamic-static load ratios. Two conclusions can be drawn from the curves of *Fig.* 5; first, the decrease in the amount of loosening for the once-used nuts as compared to the virgin specimens is very marked and, secondly, the amount of loosening is less for the lower dynamic-static load ratio (0.80) than for the higher (0.90). It thus seems evident that previous exposure of contact surfaces to cycles of alternating loading is beneficial rather than harmful and of considerable assistance in reducing the amount of loosening. The data also indicate (comparing *Figs.* 2 and 3 with *Figs.* 6 and 7) that the scatter, too, is considerably reduced. For sake of clarity, graphs in *Figs.* 6 and 7 are shown for only 4 specimens from each test group.

To throw further light on the effect of surface condition, a second series of tests, known as Series *B*, were run in which each of 20 separate 5/16-18 zinc-plated specimens was first carefully cleaned with a commercial solvent. In addition, the chamfer and face of each of the twenty nuts were smoothed to the same degree by means of a 600-grit emery paper. These additional precautions were taken in an effort to determine whether the inconsistency in the loosening curves might be caused by dust, grease or rough spots on the contacting surfaces. With this new set of specimens, the 400/500 and 450/500 tests were repeated with ten specimens (test group No. 4) and nine specimens (test group No. 5), respectively.

When the resultant data were plotted, it was found that the individual loosening curves for each of the load ratios still did not coincide but differed appreciably one from the other, although each group, as a whole, exhibited a family of curves of the same general characteristics. Upon averaging these families, however, it was discovered that the total loosening was approximately one-half that observed in the first series of tests. This is shown for Test No. 5 by the upper two curves of *Fig.* 8 for the case of dynamic-static load ratio of 450/500 or 0.9. Another difference seems to be that the rate of loosening for the cleaned, smoothed specimens changes more abruptly, in such a manner that essentially no additional loosening occurs after 4000 cycles. Similar general conclusions were obtained for the specimens of Test No. 4.

Several additional experiments were carried out to clarify the role played by condition of contact surfaces. One series of experiments, noted in TABLE 1 as Series *E* consisted of substituting rolled-thread heat-treated bolts for the standard cadmium or zinc-plated cut-thread bolts and testing these bolts on regular cadmium-plated nuts. In Test No. 15, eight specimens were subjected to a dynamic load of 600 lb and a static load of 750 lb for a dynamic-static ratio of

Fig. 2—Typical loosening curves for cut threads, cadmium plated, with a dynamic-static load ratio of 450/500 = 0.90 (Series A, Test 3)

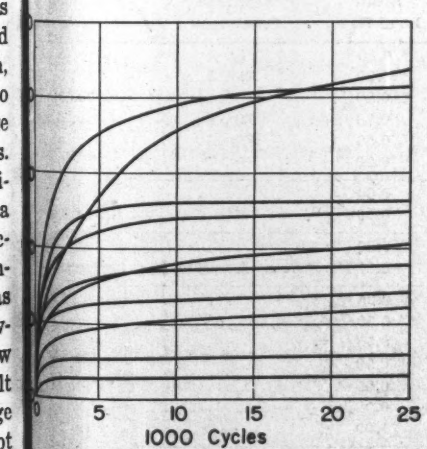


Fig. 3—Loosening curves for cut threads, cadmium-plated, with a dynamic-static load ratio of 400/500 = 0.80 (Series A, Test 2) for ten specimens

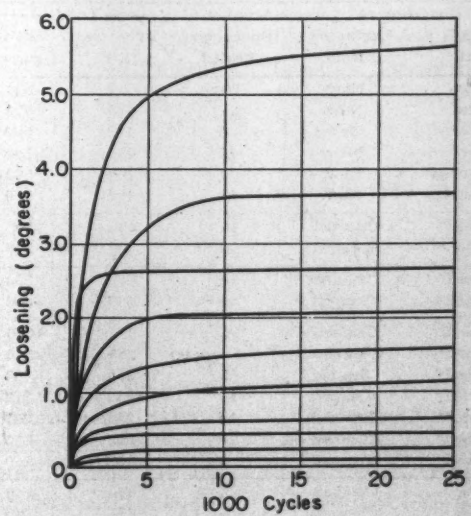
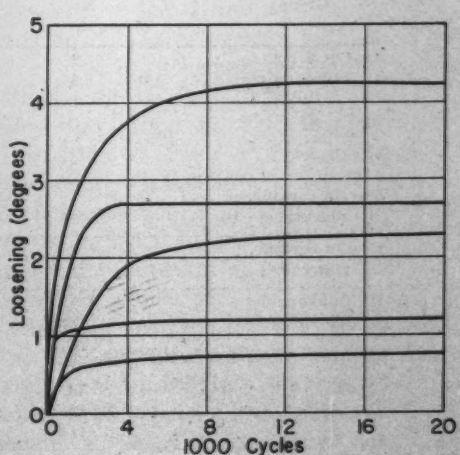


Fig. 4—Family of loosening curves for dynamic-static load ratios of 450/500 = 0.90 in which bolt rotation was checked against nut loosening (Series A, Test 3a)



0.8; in another group, Test No. 16, four specimens were subjected to the same static load by a dynamic load of 675 lb, thus providing a dynamic-static load ratio of 0.90. Some of the test results are plotted in *Figs.* 9 and 10 from which it may be seen that the amount of loosening has been considerably reduced and is even less than for the once-used nut tests of *Fig.* 5. These figures are not strictly comparable as, although dynamic-static load ratios are maintained, the load values themselves differ. Furthermore, *Fig.* 5 shows average loosening curves while *Figs.* 9 and 10 show individual loosening curves.

These various tests indicate that the amount of loosening can be reduced in several ways, such as by previous subjection of the mating surfaces to alternating cycles of loading, by subjecting the contact surfaces to a cleansing and smoothing process designed to remove dirt, grease and surface irregularities, or by use of bolts having initially smoother and more regular surfaces by reason of their method of manufacture. All of these methods are probably alike in one respect, viz, they permit greater and more intimate surface contact between the mating parts and in so doing probably result in a considerable increase of the "effective" coefficient of friction.

ALIGNMENT: Since neither previous subjection to alternating loading nor a uniform process of cleansing and smoothing completely eliminated the scatter of results between individual specimens (although both were beneficial in this respect), other causes were considered. One such cause, it was felt, might be misalignment, caused by variations in angle between head and shank of bolt and between base of nut and nut axis or by slight differences in technique of setting up specimens. In an effort to control this factor, spherical seats were constructed and installed.

Another set of specimens, Series *C*, were prepared for testing with spherical seats. Each specimen was cleaned and its nut chamfer and face smoothed as previously indicated. The usual 400/500 and the 450/500 tests were then repeated, with ten separate specimens, Test No. 6, being tested at the 0.8 dynamic-static ratio and six separate specimens, Test No. 7, at the 0.9 ratio. The experimental data when plotted again showed two families of curves and hence scatter was still present. This persistence of scatter despite the additional precautions that were taken in an effort to obtain alignment uniformity, seem to indicate that perhaps surface roughness of the threads is alone responsible for the discrepancies. The commercial manufacturing processes for standard fastenings cannot produce individual nuts with identical thread surfaces and hence it is perhaps too much to expect that consistent loosening data can be obtained on supposedly identical samples even when other sources of difference are eliminated.

Spherical Seats Improve Holding

However, when average rather than individual loosening curves are plotted, there is a significant decrease in the amount of loosening. This is clearly shown, *Fig.* 8, by comparing the average loosening data for Series *C* specimens (spherically seated), with the average results of Series *B* specimens.

TIGHTENING TORQUE AND INITIAL BOLT TENSION: Since in many applications the bolted assembly is subject to a definite and sometimes determinable alternating loading, it was thought worthwhile to run one series of tests in which the magnitude of the alternating loading was fixed, but in which the initial tightening of the nut-bolt combination and hence the initial bolt tension was varied from group to group. As noted in TABLE 1, this series of tests, Series *D*, contained seven groups (Test Nos. 8-14) of specimens, each test containing from five to ten separate specimens. This series of tests was run using the spherically seated fixtures and, prior to each test, the threads were cleaned and the nut smoothed.

It was desired that the testing procedure would simulate as closely as possible the tightening operation that would be encountered in actual practice,

TABLE 1—Summary of Loosening Tests (5/16—18 NC Class 3)

Series Group	Test No.	Condition of Threads and Surfaces† Nut	Bolt	Number of Specimens Tested	Load Dynamic (lb)	Load Static (lb)	Dynamic-Static Load Ratio	Loosening Curves Separate Specimens	Loosening Curves Specimen Averages
A—Cut threads, cadmium plated	2	New	New	20	400	500	0.80	*Fig.* 3	*Fig.* 5
	3	New	New	17	450	500	0.90	*Fig.* 2	*Fig.* 5
	2*a*	New	New	5	400	500	0.80	.....	*Fig.* 5
	3*a*	New	New	5	450	500	0.90	*Fig.* 4	*Fig.* 5
	2*b*	Once-used	New	9	400	500	0.80	*Fig.* 6	*Fig.* 5
	3*b*	Once-used	New	11	450	500	0.90	*Fig.* 7	*Fig.* 5
B—Cut threads, zinc plated	4	Cleaned*	Cleaned	10	400	500	0.80	.....	.....
	5	Cleaned*	Cleaned	9	450	500	0.90	.....	*Fig.* 8
C—Cut threads, zinc plated	6	Cleaned*	Cleaned	10	400	500	0.80	.....	.....
	7	Cleaned*	Cleaned	6	450	500	0.90	.....	*Fig.* 8
D—Cut threads, zinc plated	8	Cleaned*	Cleaned	6	250	1000	0.25	.....	*Fig.* 11
	9	Cleaned*	Cleaned	6	250	500	0.50	.....	*Fig.* 11
	10	Cleaned*	Cleaned	5	250	333	0.75	.....	*Fig.* 11
	11	Cleaned*	Cleaned	10	250	278	0.90	.....	*Fig.* 11
	12	Cleaned*	Cleaned	10	250	263	0.95	.....	*Fig.* 11
	13	Cleaned*	Cleaned	10	250	255	0.98	.....	*Fig.* 11
	14	Cleaned*	Cleaned	6	250	250	1.00	.....	*Fig.* 11
E—Nuts cadmium plated Bolts rolled threads, heat-treated	15	New	Cleaned	8	600	750	0.80	*Fig.* 9	.....
	16	New	Cleaned	4	675	750	0.90	*Fig.* 10	.....

* Nuts cleaned with faces and chamfers smoothed.

† Seating Surfaces: Series *A, B, E*—Flat plates; Series *C, D*—Spherical seats.

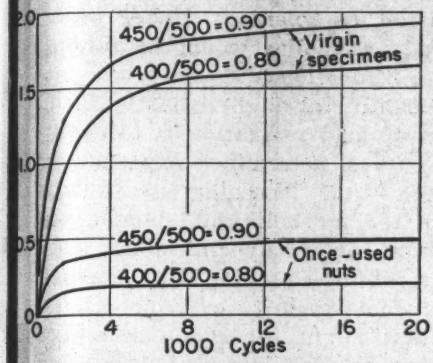


Fig. 5—Average loosening curves for Series A tests showing effects of dynamic-static load ratios and repeated use of nuts (Series A tests)

Fig. 6—Typical loosening curves for once-used nuts on new bolts with a dynamic-static load ratio of 400/500=0.80 (Series A, Test 2b)

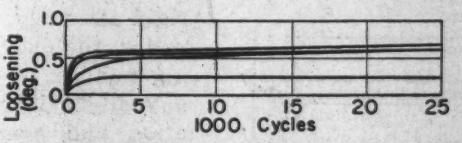


Fig. 7—Loosening curves for dynamic-static load ratio of 450/500=0.90 with once-used nuts on new bolts (Series A, Test 3b) indicate reduced scatter

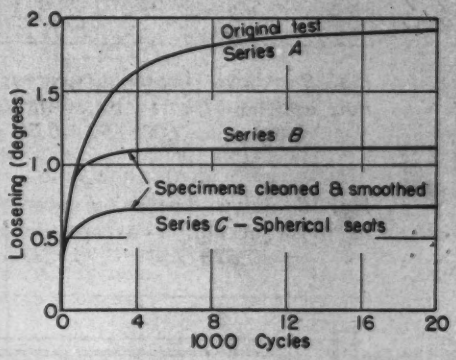


Fig. 8—Average loosening curves showing effect of contact surface conditions and effect of alignment. Dynamic-static load ratio: 450/500=0.90

assuming, say, that the nut-bolt combination were used to draw together and clamp a rigid steel assembly. Because of inherent differences in the test setup and the assumed rigid assembly of practice, this could not be done exactly. Compare, for instance, the spring constant of a bolt clamping rigid steel plates with that of the experimental setup of *Fig.* 1. In the first case, the spring constant is $K=E(A/L)=$ 650,000 lb/in. (E=modulus of elasticity of steel, A=cross-sectional area of bolt body, and L=length of bolt under tension) while in the experimental apparatus it is the same as that of the preload springs, viz. 3150 lb/in.

Simulated Tightening Conditions

The assumed practical conditions were approximately simulated by turning the torque wrench only through that angle which would give the same relative rotary motion of the nut-bolt combination that it would experience if clamping rigid plates. In other words, before the tightening torque was applied, the bolt-nut specimen was preloaded by the preload springs to such a point that the rotary motion and deflection accompanying the subsequent tightening brought the stress to the desired initial static value.

In all Series *D* tests, the sinusoidal alternating load was maintained at a magnitude of 250 lb, while the static load measured by deflection of the preload springs was varied from 1000 lb in Test No. 8 to 250 lb in Test No. 14. This corresponded to a range of dynamic-static ratios from 250/1000=0.25 to 250/250 =1.00. The test data, shown in the form of average loosening curves for each of seven different values of initial bolt tension varying from 1000 to 250, are shown in *Fig.* 11. These graphs illustrate the effect of dynamic-static load ratio, confirming the previous indications that loosening increases with increase of this ratio. There are some slight abnormalities, such as the 0.90 curve showing slightly greater loosening than the 0.95 curve but such differences are probably to be expected in view of the scatter in the individual loosening curves and the limited number of specimens that were tested in each group. *Fig.* 11 also shows the trend of loosening as a function of tightening torque even though for any given group of specimens individual values of tightening torque varied widely, *Fig.* 12. However, it is seen from this figure that if average values of the tightening torques for each group of specimens are plotted as a function of bolt tension, a linear relation is found over the range tested. Therefore, for a given dynamic loading, the amount of loosening will decrease with increase of tightening torque and hence the importance of proper tightening, so as to keep the dynamic-static load ratio small, cannot be overstressed.

LOAD RELAXATION: If loosening occurs, no matter how little, when a mechanical assembly is subject to vibration, the bolt tension will be somewhat reduced and the clamping force will be somewhat relaxed. This, in turn, will produce an increase of the dynamic-static load ratio and hence, as *Figs.* 5 and 11 show, there will be a tendency for the loosening to increase. We have also seen, however, (*Figs.* 2 and 3) that the rate of loosening tends to decrease rapidly with increase in number of cycles of alternating loading. Hence, in practice, there will be two opposing tendencies operating once loosening under vibratory loading has started; one, due to effect of load cycles tends to retard the loosening forces, the other, due to load relaxation and consequent increase in dynamic-static load ratio, tends to increase loosening.

The importance of load relaxation is masked in the experimental apparatus since the soft springs used to apply the static load permit virtually no relaxation to take place. A loosening of one degree, for example, corresponds to a relaxation of only 0.5-lb in the bolt load. Had the bolt been used, however, to clamp a rigid steel assembly, a loosening of one degree would correspond to 100 lb load relaxation. The data of *Fig.* 11 may now be used to obtain, for any given number of load cycles, "equivalent" load

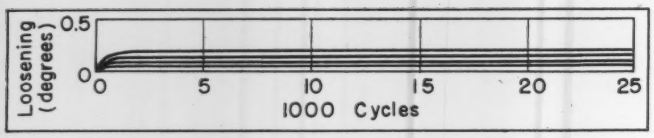
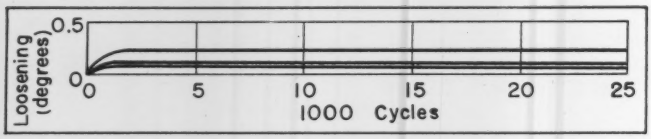


Fig. 9—Above—Loosening curves using cadmium-plated nuts and heat-treated rolled-thread bolts. Dynamic-static load ratio: 600/750 = 0.80 (Series E, Test 15)

Fig. 10—Below—Loosening curves for nuts and bolts similar to those in Fig. 9 except dynamic-static load ratio is 675/750 = 0.90 (Series E, Test 16)



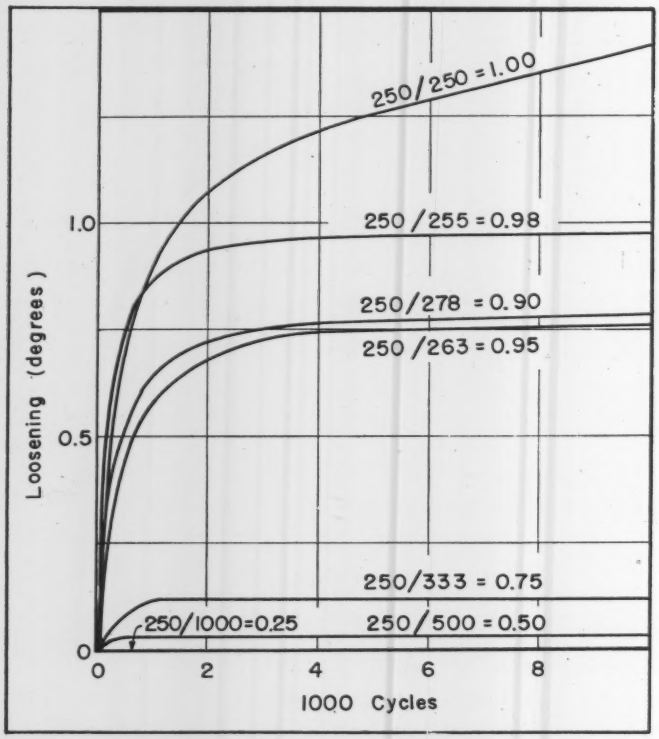


Fig. 11—Average loosening curves for cut threads, zinc plated, which have been cleaned and nut faces smoothed. Effect of initial bolt tension and effect of dynamic-static load ratio are indicated (Series D tests)

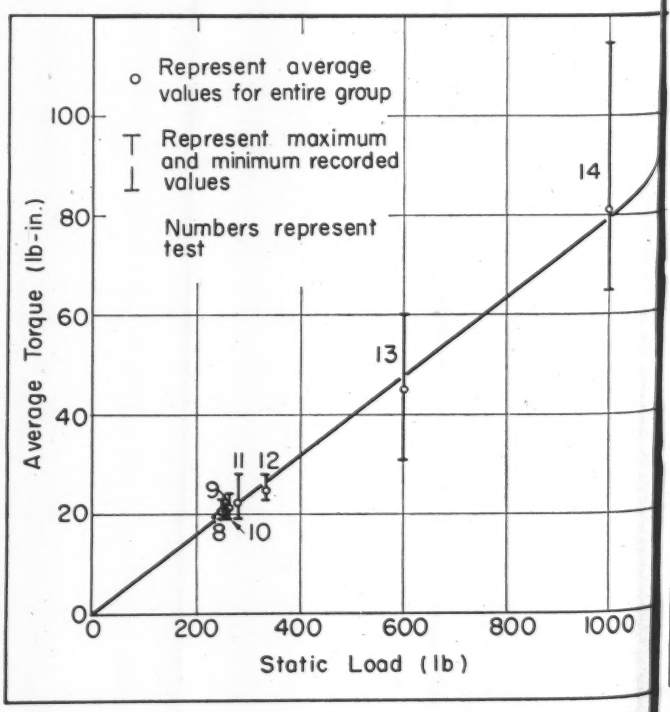
relaxation as a function of dynamic-static load ratio (or as a function of initial bolt tension or tightening torque, whichever is desired).

One such equivalent load relaxation curve for specimens of Series *D* for 10,000 cycles of dynamic stressing is shown in *Fig.* 13. In this figure, equivalent load relaxations are also shown for specimens of Series *C*, corresponding to the average loosening curves for dynamic-static load ratios of 0.80 and 0.90. Even though the actual dynamic-static load magnitudes differed in these two series of tests, it is interesting to note that the plotted points all follow the same trend. There appears to be little load relaxation for low dynamic-static ratios but the curve exhibits a "knee" in the neighborhood of a dynamic-static load ratio of 0.8. For higher ratios, the amount of load relaxation becomes a significant percentage of the original static load and hence dynamic-static load ratios greater than 0.8 should definitely be avoided in practice. Load relaxations of 10 or 20 per cent or more usually can not be tolerated. Under these conditions most bolted assemblies would be failing to fulfill their primary function.

Although felt particularly at high dynamic-static load ratios, the effect of load relaxation is to cause the actual loosening curves, determined experimentally, to differ from the "true" loosening curves that would be found were the nut-bolt unit clamping a rigid assembly. However, for low dynamic-static load ratios, the decrease in rate of loosening with increasing number of cycles would seem to overcompensate any slight increase that might occur due to relaxation and hence the corresponding true loosening curves should be little different from those of *Figs* 8 or 11. Probably the only effect would be to slightly shift the relatively sharp knee of *Fig.* 13 to a somewhat lower ratio, say, 0.7. Use of higher ratios in actual practice would then be unwise, whereas dynamic-static load ratios of 0.60 or less would seem safe.

The experimental data indicates the effect of a number of specific variables on loosening. It should be borne in mind, however, that all tests were run at one thread helix angle on one size specimen (5/16-18), and at one frequency of alternating vibration (30 cps). It would be well before generalizations from these test results are made to have additional test information on the loosening of specimens of different size, different helix angle, and subject to different frequencies of vibration. Unfortunately, very little such data are available. The only other test information of which we are aware is that of Goodier and Sweeney[1], who conducted tests in some ¾-inch diameter American Standard nuts having

Fig. 12—Tightening torques plotted against static load showing linear relationship (Series D tests)

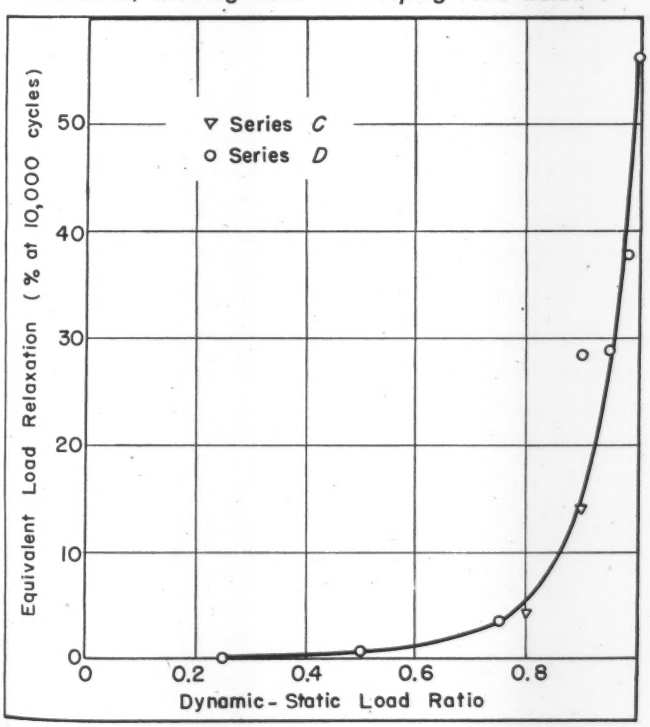


10 threads to the inch. They applied alternating loading at very low frequencies by setting up the specimens in a standard type tensile machine and then loading and unloading for some 500 cycles. They measured nut-bolt rotation during both the loading and unloading portions of a load cycle and found the cumulative loosening to vary with the type of machine used (total of three) and from specimen to specimen for a given machine. Since their vibratory loading was not continued past 500 cycles, it is hardly possible to note the large reduction in rate of loosening found in these high-frequency tests. However, their reported rates of loosening do appear to decrease with increase in the number of load cycles.

The various phenomena that have been observed in these tests seem to be intimately connected with the basic nature, and the relative smoothness and fit of the contacting surfaces. For example, variations of surface conditions from specimen to specimen are unavoidable in the manufacture of threaded parts. Hence, if surface conditions do play a prominent part in the loosening process, it would be expected that the amount of loosening would vary from specimen to specimen even when other factors such as dynamic-static load ratio, alignment, etc. were kept constant.

Reduction in rate of loosening with increase of load cycles is probably directly attributable to changes in surface conditions produced by action of the dynamic load and might reasonably be described in terms of an effective coefficient of friction μ' which itself increases with number of vibrations. When μ' reaches some critical value, loosening stops and the rate of loosening tends to approach zero as indicated by the curves of *Figs.* 2 and 3. Reduction of loosening by cleansing or smoothing the surfaces with sandpaper is then to be expected as both of these processes would tend to increase the surface coefficient of friction and hence the overall frictional resistance to rotation. Improvement of alignment, resulting in greater seating contact and more intimate mating of the contacting parts likewise would be expected to increase the effective coefficient of friction.

Fig. 13—Effect of dynamic-static load ratio on static load relaxation, showing need for keeping ratio below 7



How To Prevent Loosening

Although existing data are limited, steps to be taken or procedures to be adopted in overcoming vibration loosening include:

1. Proper specification of working loads
2. Smooth, clean contact surfaces
3. Alignment of contacting parts.

The most important would appear to be proper specification of working loads so that the dynamic load will not approach or exceed the static tensile load originally developed in the tightening operation. For a given magnitude of dynamic load, this would imply proper control of the tightening operation so as to insure the building up of a static tension greater in value than the dynamic load to be carried. Exactly how much greater is difficult to say, but the results of the present investigation indicate that it is desirable to keep the dynamic-static load ratio in the range of 0.7 or less if the deleterious effects of load relaxation are to be overcome. As has been strongly emphasized by Almen[6], there is no question but that the "man with the wrench" carries responsibility.

Another significant precaution is adequate control of the method of manufacture, fabrication, assembly, and maintenance so as to have all contact surfaces at the threads, base of nut and head of bolt, in the best possible condition of cleanliness and smoothness. Smooth, clean surfaces provide higher frictional resistance to rotary motion and are helpful in reducing the amount of loosening which tends to take place under vibratory loading.

Improvement in alignment is also helpful as it results in better mating of the contacting parts in somewhat the same manner as does removal of dirt, grease or surface irregularities. The use of high friction coefficient washers between the face of the nut and the seating surfaces, as advocated by Sweeney[7] is but another way of increasing the so-called effective coefficient of friction and, as such, will be beneficial. In general, any steps leading to better mating of parts and to greater uniformity of the degree of smoothness and cleanliness of the surfaces—such as, for example, the simple subjection of the mating surfaces to cycles of dynamic loading—will be of assistance in overcoming loosening.

References

1. J. N. Goodier and R. J. Sweeney—"Loosening by Vibration of Threaded Fastenings", *Mechanical Engineering*, Dec. 1945.
2. B. J. Lazan—"Fatigue Testing Machine", MACHINE DESIGN, May 1947.
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6. J. O. Almen—"On the Strength of Highly Stressed, Dynamically Loaded Bolts and Studs", *Society of Automotive Engineers Journal* (Trans.) April 1944.
7. R. J. Sweeney—"How to Avoid Loosening of Bolts and Nuts," MACHINE DESIGN, June 1946.

MILESTONES IN DESIGN

By John Kremer
Cleveland, O.

The Dictating Machine

SOME time after man first began to use audible speech he wanted to be free of the limitations of his own memory and mouth for the retention and voicing of words. Out of this grew the various phonetic alphabets and idiographs and scripts and one of the chief avenues along which he progressed until 1880. That was by writing and reading ". . . and of the making of many books there is no end."

But there were other approaches. The powerful had slaves to memorize and repeat; and there were wandering minstrels in medieval times. Those who had a way with birds trained them to talk, albeit capriciously: Pliny in the 1st Century A. D. remarks of the parrot, "India sends us this bird—it will duly salute an emperor and pronounce words it has heard; it is especially frolicsome under the influence of wine." Those of philosophic turn realized that no living thing was needed for the storage and later repetition of sounds as echoes.

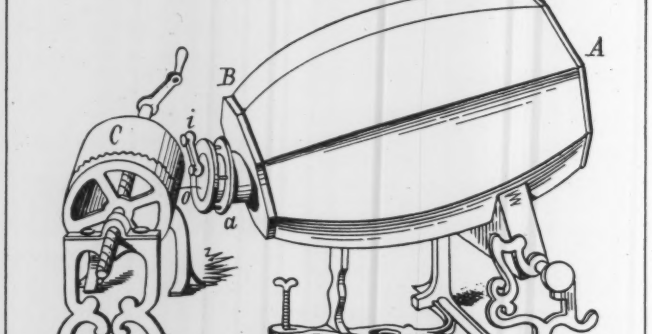


Fig. 1—Left—Phonautograph devised by Scott in 1857 and constructed by Koenig. Marking stylus was attached to a sensitive diaphragm

Fig. 2—Below—An early sound record made by an instrument similar to the Phonautograph in Fig. 1

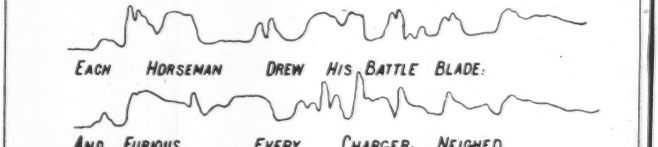
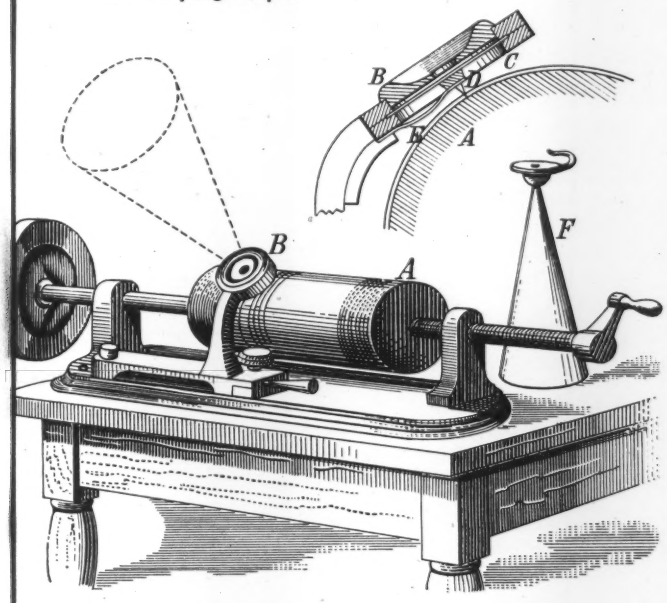

By Torch And Trumpet Fast Arrayed,
Each Horseman Drew His Battle Blade:
And Furious Every Charger Neighed,
To Join The Dreadful Revelry.

Fig. 3—Below—Edison's cylinder type Phonograph, patented in 1878, which employed a groove of varying depth



200,521. PHONOGRAPH OR SPEAKING MACHINES. T. A. Edison, Menlo Park, N. J. Filed Dec. 24, 1877.

To all whom it may concern:

Be it known that I, THOMAS A. EDISON, of Menlo Park, in the county of Middlesex and State of New Jersey, have invented an Improvement in Phonograph or Speaking Machines, of which the following is a specification:

The object of this invention is to record in permanent characters the human voice and

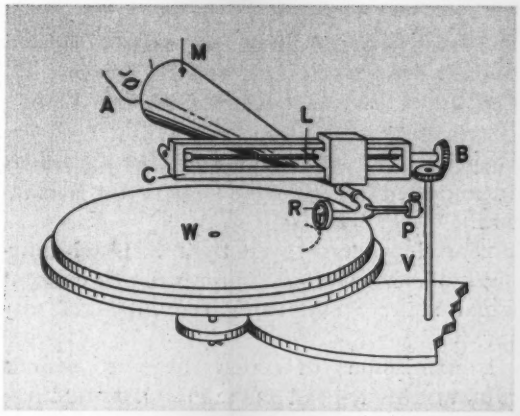
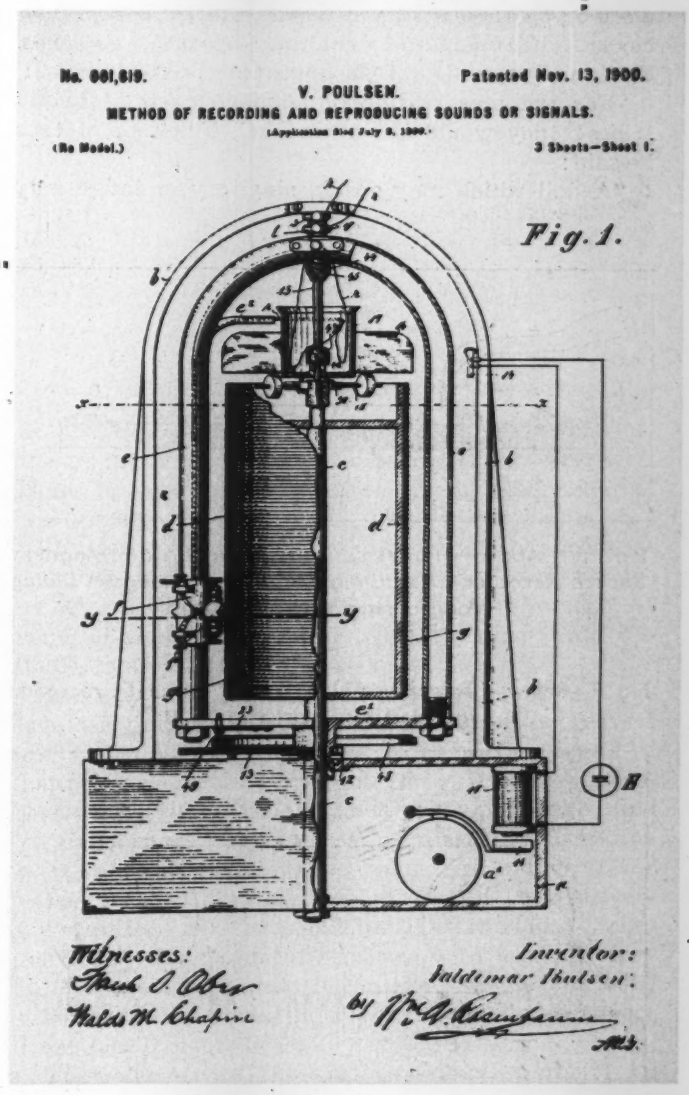


Fig. 4—Above—Edison's wax disk type Phonograph recorder in which groove was sinuous with approximately uniform depth and width

Fig. 5—Below—Patent showing Poulsen's first disclosure, in this country, of a magnetic recorder for "reproducing sounds or signals"


No. 661,619. Patented Nov. 13, 1900.

V. POULSEN.

METHOD OF RECORDING AND REPRODUCING SOUNDS OR SIGNALS.

(Application filed July 8, 1900.)

(No Model.) 3 Sheets—Sheet 1.

Fig. 1.

Witnesses:
Frank P. Ober
Waldo M. Chapin

Inventor:
Valdemar Poulsen,
by Wm. A. Simpson
Atty.

In the last quarter of the 19th Century Thomas A. Edison, who in 1869 at the age of 22 had received $40,000 for his stock-ticker printing telegraph invention, was seeking new fields to conquer. He was a keen student of Faraday and other great European scientists and undoubtedly knew of the work of Leon Scott and Koenig, announced in 1857, on the Phonautograph, *Fig.* 1, that caused sound to inscribe itself as marks of a whisker attached to a diaphragm at one end and bearing on a moving smoked surface at the other, *Fig.* 2.

Edison, who was a leader in improving the telegraph and had been working on speaking tubes and the carbon-button and diaphragm telephone transmitter, was granted U. S. patent 200,521 on his Phonograph, *Figs.* 3 and 4, in 1878.

In Edison's talking Phonograph patent he spoke of ". . . a diaphragm capable of being vibrated by the human voice in conjunction with a material capable of registering the movement of such vibrating body . . . by altering such material in such a manner . . . that will be sufficient to cause a second vibrating body to be set in motion and reproduce the motion of the first vibrating body." He mentioned a foil record. In the days before the development of modern plastics it was, perforce, a metallic foil. In

Edison's first patent he spoke of the hill and dale and the lateral-cut record and of the electrically actuated recorder.

Here are some extracts from an article Edison wrote in 1878 on the phonograph and its future. He announced as novel accomplishments of the Phonograph:

"1. The captivity of all manner of sound-waves heretofore designated as 'fugitive' and their permanent retention.

"2. Their reproduction with all their original characteristics at will, without the presence or consent of the original source, and after the lapse of any period of time.

"3. The transmission of such captive sounds through ordinary channels of commercial intercourse and trade in material form, for the purpose of communication or as merchantable goods. (The mailable record was to replace the written letter.)

"4. Indefinite multiplication and preservation of such sounds, without regard to the existence or nonexistence of the original source.

"5. The captivation of sounds, with or without the knowledge or consent of the source of their origin."

And as more important applications he mentioned:

"Letter-writing and other forms of dictation, books, education, reader, music, family-record; and such electrotype applications as books, musical boxes, toys, clocks, advertising and signalling apparatus, speeches, etc.", and says also of phonographic books that:

"For the preservation of languages (as of Indian tribes) they would be invaluable." Writing of toys he said:

"A doll which may speak, sing, cry or laugh may be safely promised our children for the Christmas holidays ensuing. Every species of animal or mechanical toy—such as locomotives, etc.—may be supplied with their natural and characteristic sounds," anticipating much of what was to come in the production of cartoons of the sound movies and radio broadcasts of half a century later.

Edison's Phonograph was, of course, the predecessor of the familiar groove type office dictating machines in use today.

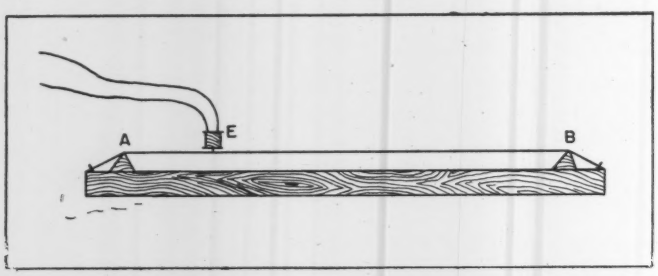
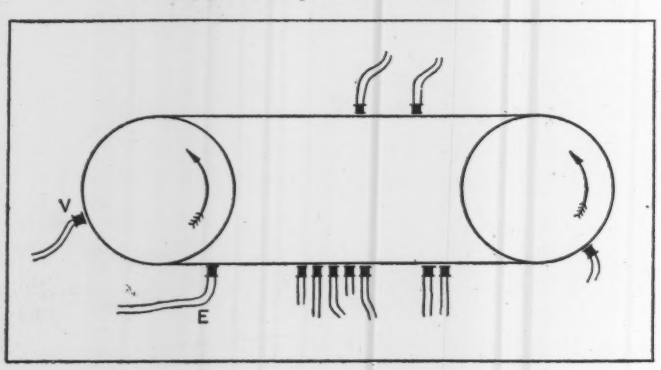


Fig. 6 — Above — Poulsen's Telegraphone, a "Magnetic Speech Recorder", consisting of an electromagnet sliding along a length of piano wire

Fig. 7—Below—"Telephonic Newspaper" magnetic recorder suggested by Poulsen in 1900



Magnetic Recorder Described in 1888

It is not generally realized that the modern magnetic tape recorder was anticipated by Oberlin Smith over 60 years ago. Writing in the *Electrical World* of September 8, 1888, Smith disclosed for the first time the principles of the magnetic dictating machine. He said:

"The voice or other sound is delivered into an ordinary telephone. The current, broken into waves of varying lengths and intensities corresponding with the vibrations of the diaphragm in the telephone, passes in its circuit through a helix, converting into a permanent magnet any piece of hardened steel which may be at the time within the helix. Through this helix passes a cord, string, thread, ribbon, chain, or wire made wholly or partly of hardened steel, and kept in motion by being wound on to the reel.

"Another imaginable form would be simply a hard steel wire, but it is scarcely possible that it would divide itself up properly into a number of short magnets. The magnetic influence would probably be distributed along the wire in a most totally depraved way, with nodal points just where they were not wanted. If it could be made to work it obviously would be the simplest thing yet suggested."

Poulsen's Magnetic Recorder Patented in 1900

But in 1900 Valdemar Poulsen made it work, *Fig.* 5. In Poulsen's words, written in that year:

"The following is a description of the principles and the arrangement of my invention, which I have called the Telegraphone. A steel wire (piano wire) *AB,* about 1.5m (5 ft) long and 0.5m (1/50-inch) in diameter is stretched on a board (*Fig.* 6). Along it can slide the electromagnet, *E,* which embraces it with one of the poles. The core of the electromagnet is a piece of soft iron wire about 8mm (1/3-inch) long and 0.75mm (3/100-inch) in diameter, and the electromagnet itself is in series with a battery and microphone, or is connected to a transformer in the microphone circuit. At the beginning of an experiment the wire should be completely unmagnetized.

"If, while the electromagnet is sliding along the wire with a velocity of about 1m per second, the microphone is spoken into, the current fluctuations produced register themselves by means of the electromagnet on the steel wire. If now the electromagnet is connected up with a telephone and made to travel over the wire again, the telephone repeats what was spoken into the microphone. Thus, owing to the great coercive strength of steel, there has been

(*Concluded on Page* 206)

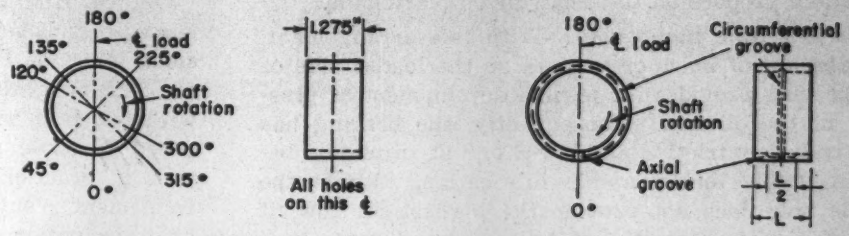


Fig. 1 — Arrangements of oil holes and grooves in bearings

Sets 31 & 51, oil hole at 0°
Sets 32 & 52, oil holes at 120° & 300°
Sets 34 & 54, oil holes at 45°, 135°, 225°, and 315°
All holes 0.125" diam., chamfered to 0.18" diam. at edge

Sets 33 & 53, axial groove, 0.10" width, 0.04" depth
Sets 35 & 55, circumferential groove, 0.150" width, 0.075" depth
Average shaft diam. – 2.055"

Effects of Oil Holes and Grooves On Bearing Performance

By S. A. McKee and H. S. White

National Bureau of Standards

Washington, D. C.

ONE of the problems in the lubrication of plain journal bearings is the method of admitting oil to the bearing. The concept of the load-carrying film indicates the desirability of avoiding the use of oil holes or grooves which interfere with the normal development of hydrostatic pressure to support the load. In some bearing installations, however, it is not always possible to satisfy this requirement. With bearings where loads are fluctuating in both intensity and direction, it is sometimes impracticable to apply the oil to the unloaded side throughout the complete load cycle. Also, in some cases provision must be made for a continuous flow of oil to some other moving part.

This article discusses tests which were made to determine the comparative performance of journal bearings with various arrangements of holes or grooves when using forced-feed lubrication under conditions where the load was unidirectional on the bearing. Operation was tested with three arrangements of oil holes and two types of oil grooves in the surface of the bearing. Also, three arrangements for feeding the oil from the center of a hollow shaft were tried. Holes or grooves were located at the axial center of the bearing. Two clearance-diameter ratios were used with each arrangement and test runs were made at constant speed. The data cover operation with one oil at one oil-inlet temperature and two speeds. In some of the tests the loads were increased until unstable lubrication was reached.

In TABLE 1, f = coefficient of friction; Z = absolute viscosity of lubricant at atmospheric pressure and bearing temperature, centipoises; N = speed of journal, revolutions per minute; P = pressure on projected area of bearing, pounds per square inch; and ZN/P = generalized operating variable. Lubrication is more stable and bearings wear less at high ZN/P values. Friction ratings are given as: *A*, relatively low friction; *B*, intermediate friction; and *C*, high friction. Since load ratings vary with bearing "run-in," values for the fourth run are used in indicating relative load-carrying capacity. Capacity decreases from *A* to *D* in the table.

The influence of the various arrangements of oil

From a paper presented at the ASME Spring Meeting, Washington, D. C., April, 1950.

holes and grooves on the behavior of bearings is:

One Oil Hole in Bearing. With this arrangement, the absence of holes or grooves on the loaded side of the bearing provides for normal development of pressure in the oil film. Consequently, the bearing has relatively low friction and low ZN/P at transition between stable and unstable lubrication. While the single hole does not provide the highest oil flow, it would seem to be sufficient to provide adequate heat dissipation for most conditions.

Two Oil Holes in Bearing. The hole on the loaded side with this arrangement apparently disturbs the normal development of pressure in the film sufficiently to make a measurable increase in friction and ZN/P at the transition point. The particular location of the oil hole on the unloaded side is adverse from the standpoint of oil flow, hence the rate of heat dissipation is somewhat lower than the arrangement with one hole in the bearing.

Axial Groove in Bearing. This arrangement apparently provides a slightly better distribution of oil for the development of the load-carrying film than the arrangement with one hole in the bearing. Consequently, the bearing has low friction and lowest critical ZN/P, and the high oil flow provided is advantageous where forced cooling is necessary.

Four Holes in Bearing. This arrangement has two holes on the loaded side, which disturbs the development of pressure in the film to a greater extent than the arrangement with two holes in the bearing; its critical ZN/P is higher, but the friction is comparable. The two holes on the unloaded side are not in advantageous positions, and the oil flow and rate of heat dissipation are comparable to the arrangement with one hole in the bearing.

Circumferential Groove in Bearing. The groove dividing the bearing into two narrower parts increases the friction and causes a relatively high critical ZN/P. The high friction is counteracted by high oil flow and rate of heat dissipation.

One Oil Hole in Shaft. The passage of one oil hole across the loaded portion of the oil film has relatively small effect on both friction and critical ZN/P, which are only slightly higher than for the arrangement with one hole in the bearing. When the hole is exposed to some portions of the loaded area, the oil feed is practically shut off. Consequently this arrangement results in the lowest oil flow and rate of heat dissipation.

Two Oil Holes in Shaft. With two oil holes for each journal, the disturbance to film-pressure development occurs twice in a revolution, resulting in slightly higher friction and critical ZN/P than with the arrangement with one hole in the shaft. The two holes also provide a greater oil flow and rate of heat dissipation. In this respect performance is comparable to the test arrangement having two holes in the bearing.

One Oil Hole with Flat On Shaft. The flat at the end of the oil hole causes considerable disturbance to the development of pressure in the oil film, increases the friction, and markedly increases the critical value of ZN/P. Oil flow and rate of heat dissipation are higher than the arrangement with one hole in the shaft without the flat, but not sufficient to counteract the effect of high ZN/P at transition from stable to unstable lubrication.

The results of these tests apply directly to unidirectionally loaded bearings for the particular range of conditions covered. The indicated differences between the various arrangements are not necessarily strictly indicative of more complex conditions where the load varies in both intensity and direction with respect to the bearings. It is believed, however, that the relative values obtained may be useful qualitatively in estimating the overall effects of various arrangements of holes or grooves under more complex loading conditions, especially if proper consideration is given to conditions present throughout the complete load cycle.

Table 1—Bearing Dimensions, Friction and Load Ratings

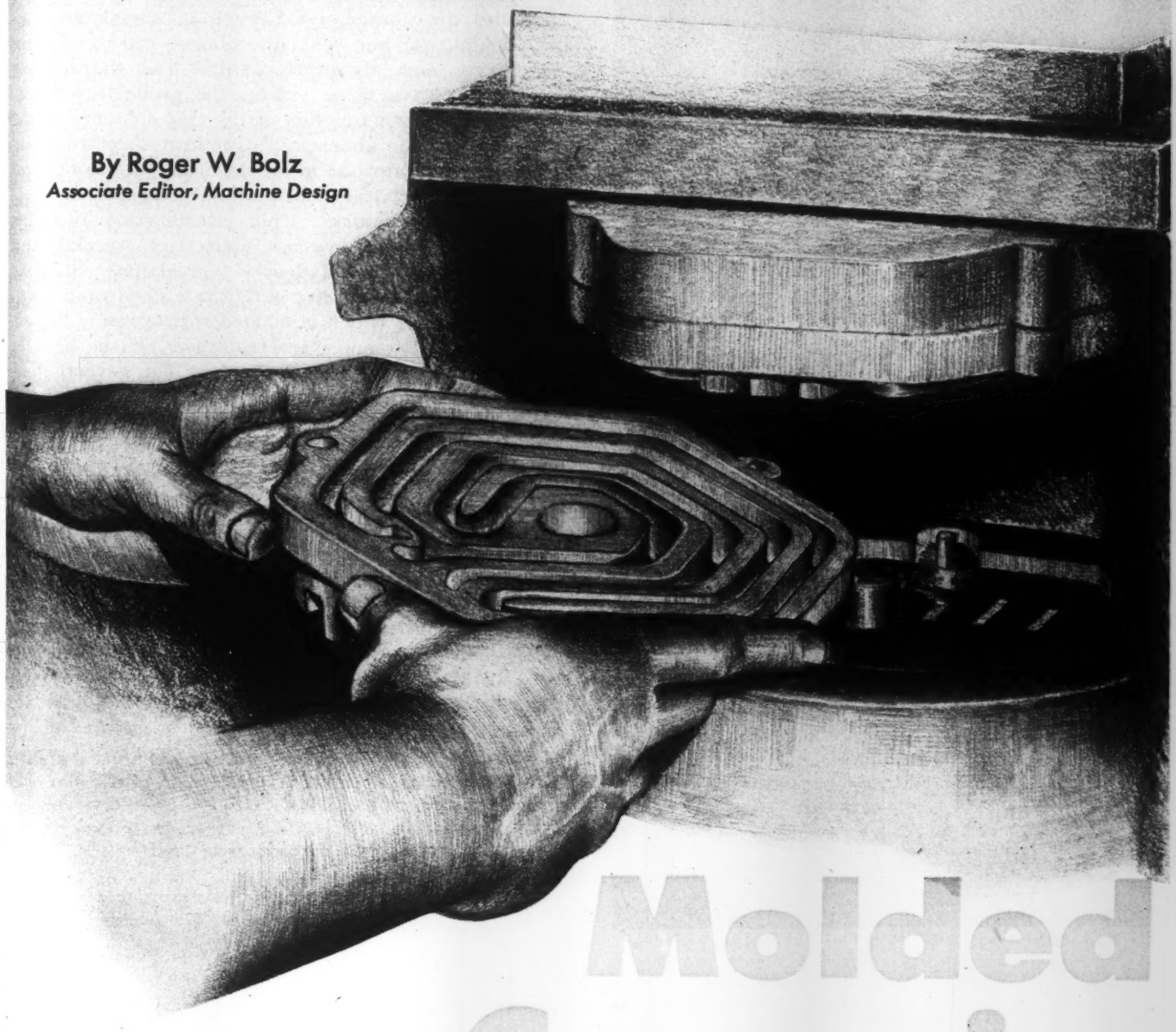
		—Shaft and Bearing Dimensions—			—Friction Ratings—				—Bearing Capacity Ratings—		
Design of oil feed	Bearing set	Avg clearance, C (in.)	Diam-clearance ratio, D/C	Length-diam ratio, L/D	$\Delta f‡$ at $ZN/P=10$	$\Delta f‡$ at $ZN/P=70$	Avg $\Delta f‡$ for each design	Friction rating	Avg critical ZN/P for each design (1st run)	Avg critical ZN/P for each design (4th run)	Load-carrying capacity rating
1 hole in bearing	31	0.0031	663	0.620	0.00101	0.00149	0.00130	A	3.2	1.6	A
	51	0.0051	403	0.620	0.00106	0.00164					
2 holes in bearing	32	0.0031	663	0.620	0.00128	0.00187	0.00147	B	3.6	2.7	B
	52	0.0051	403	0.620	0.00113	0.00159					
Axial groove in bearing	33	0.0031	663	0.620	0.00101	0.00143	0.00124	A	2.9	1.4	A
	53	0.0051	403	0.620	0.00101	0.00149					
4 holes in bearing	34	0.0031	663	0.620	0.00125	0.00165	0.00143	B	4.4	3.4	C
	54	0.0051	403	0.620	0.00116	0.00166					
Circum. groove in bearing	35	0.0031	663	0.547*	0.00186	0.00340	0.00240	C	4.2	2.8	B
	55	0.0051	403	0.547*	0.00149	0.00284					
1 hole in shaft	S31	0.0031	663	0.620	0.00109	0.00166	0.00136	A or B	2.6	1.8	A
	S51	0.0051	403	0.620	0.00107	0.00162					
2 holes in shaft	S32	0.0031	663	0.620	0.00112	0.00169	0.00143	B	2.8	1.9	A
	S52	0.0051	403	0.620	0.00111	0.00181					
1 hole with flat in shaft	S33	0.0031	663	0.620	0.00137	0.00153	0.00146	B	7.4	5.4	D
	S53	0.0051	403	0.620	0.00126	0.00166					

* Allowance made for circumferential groove. ‡ L/D correction representing difference between experimental data and Petroff equation.

PRODUCTION PROCESSES

THEIR INFLUENCE ON DESIGN • PART 53

By Roger W. Bolz
Associate Editor, Machine Design

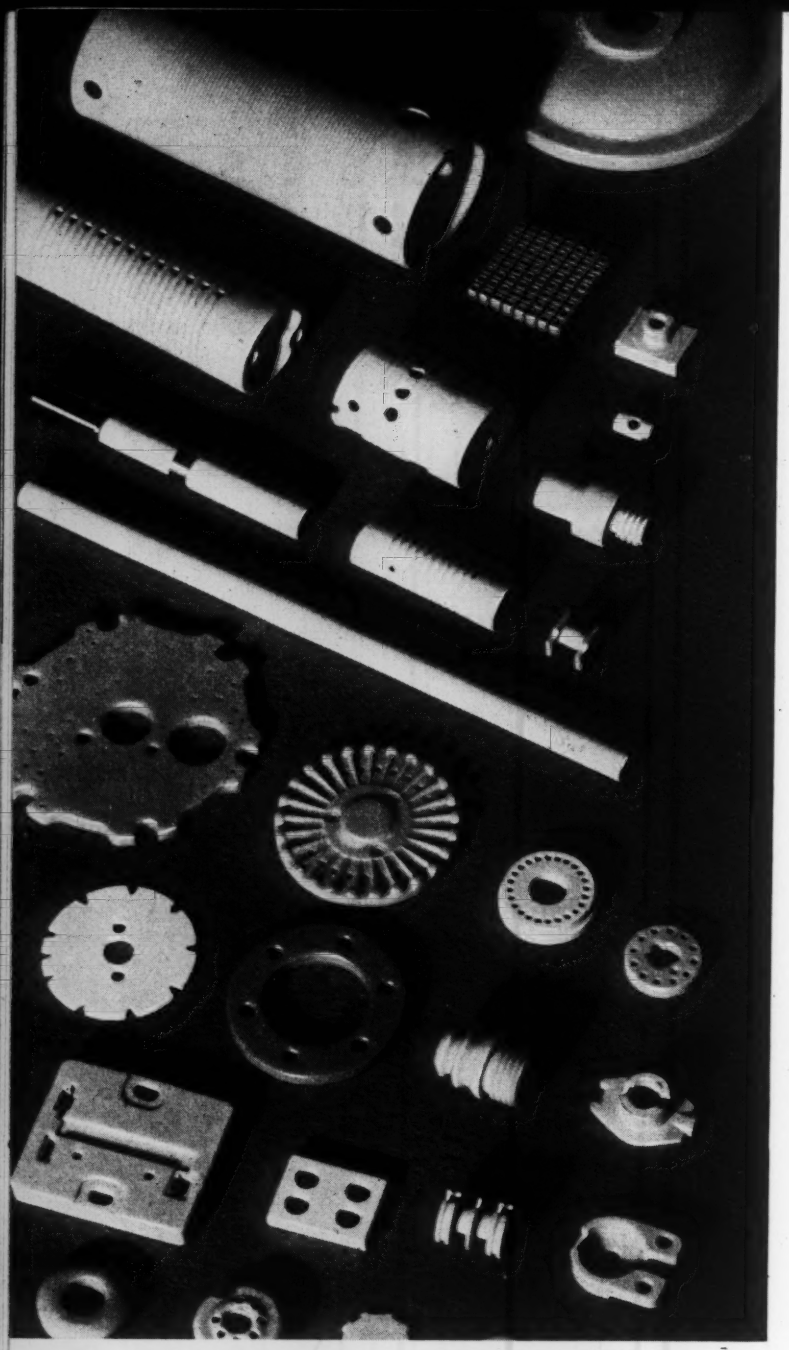


Molded Ceramics

AMONG the earliest objects fashioned by man, ceramic products have withstood the test of centuries and still find application in numerous fields. Exceptional properties of ceramics have increased their use and their availability and have brought about the development of highly specialized technical ceramics suited to unusual industrial applications, *Fig.* 1. Properly chosen and suitably designed, technical ceramics can fill a real need in the design of industrial equipment.

Failure to consider the use of ceramics in mechanical equipment generally stems from the rather meager knowledge of these materials and lack of data with which to design such parts to assure not only better performance but lower production costs.

CERAMICS: Traditionally, ceramic materials have been composed largely of naturally occurring clays, alone or in admixture with various amounts of quartz, feldspar and other nonmetallic minerals. The term ceramic often implies, in addition, the field of articles made from glass but in this article glasses will not be considered owing to the difference in manufacture.



High-Clay Ceramics: In all probability, parts produced from compositions consisting predominantly of clays account for the largest tonnage of manufactured ceramics. For convenience, these may be said to have a clay content in excess of 50 per cent. These ceramics are characterized by high shrinkage—approximately 25 per cent by volume—during drying and firing and, therefore, exhibit the widest size variation. To a large extent, the production tolerances are a direct function of the clay content. Parts produced include almost all structural clay products, clay-base refractories, most chemical porcelains and stoneware, electrical and mechanical porcelains.

Low-Clay Ceramics: This classification includes steatite and other low-loss dielectrics, special and super-refractories, and special porcelains. Steatite normally carries over 80 per cent talc (hydrated magnesium silicate) which is bonded with ceramic fluxes. Shrinkage is low and close tolerances can be held.

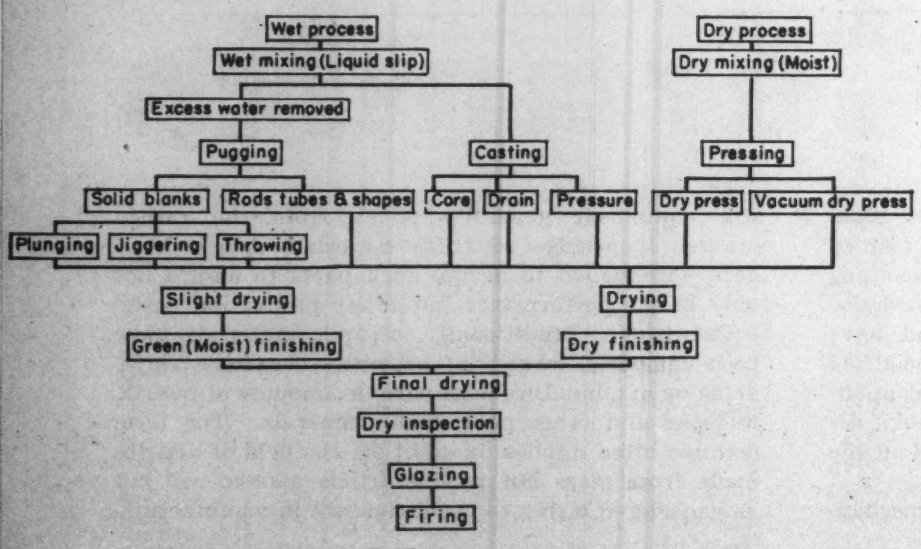
Clay-Free Ceramics: This small but nevertheless important group of ceramics has compositions entirely devoid of clay. Included in the group are the pure oxide types and the so-called cermets which have properties between metals and ceramics.

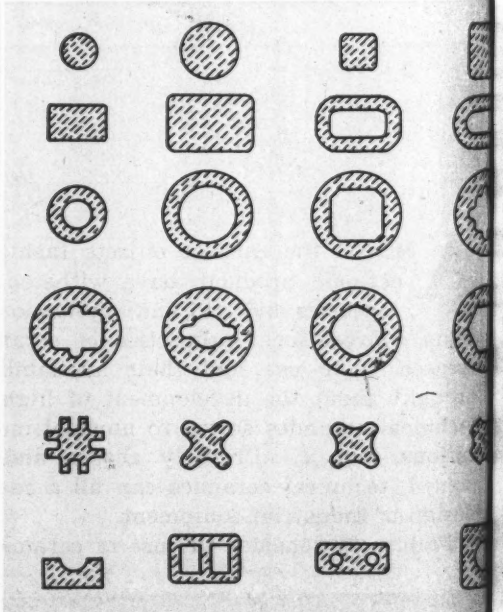
PRODUCTION METHODS: Second most important effect on the tolerances to which a ceramic part can be manufactured is produced by the method of forming involved. Various methods are used for forming ceramic bodies prior to firing and consequently practical production design must take cognizance of not only the tolerance effect of the method required but also the general range and limitations imposed. Categorically speaking the various forming methods are

Fig. 1—Left—A variety of intricate ceramic parts designed for industrial, electronic, and machine applications

Fig. 2—Left, below—Chart of ceramics manufacturing processes and procedures utilized today

Fig. 3—Below—Group of cross sections showing possibilities of extrusion process for uniform sections





either of the wet or the dry process type, *Fig.* 2.

Ceramic to be formed by one of the wet methods is given the desired plasticity through the use of a suitable quantity of plastic clay together with sufficient water to provide a relatively soft, plastic mass. When the physical requirements of the fired material make it impossible to secure the required plasticity by use of clay and water, the mix may be plasticized by means of organic binders and plasticizing agents. With low-clay and clay-free ceramics, however, the dry process is often used, semimoist granular powder having 2 to 12 per cent water by weight being compressed in dies to the desired form. Ceramics to be molded by casting are made fluid by the addition of defloculants such as sodium silicate, sodium carbonate, or solutions of salts which yield hydroxyl iron and do not ordinarily contain any more water than those prepared for ordinary wet methods.

PLASTIC WET PROCESSES: Oldest and probably most diversified general process, plastic wet forming consists of five methods of preliminary shaping—extrusion, throwing, jiggering, pressing, and casting. Method of preliminary forming to be used depends upon the part design features and the economical manufacturing considerations.

Machining Done Before Firing

After preliminary forming, the molded pieces are subjected to a drying period under controlled heat and humidity. Degree of drying depends upon the dimensional accuracy desired in the finished part. Any final shaping required is performed at this stage and, if glazing is necessary, the piece is dipped or sprayed with a mixture of glazing materials suspended in a relatively high percentage of water and again dried. To develop the final physical, electrical and mechanical characteristics the formed pieces are fired in a kiln at approximately 2200 F.

Extrusion: Most frequently used extruder is the auger or screw type, usually fitted with a vacuum chamber for removal of occluded gases. Often referred to as pugging, extrusion may be done with or without a mandrel or core to shape sections of almost any variety, *Fig.* 3.

Although this method is limited to pieces of uniform cross section, machining prior to firing may be used to produce the desired external shape, *Fig.* 4. While internal machining can be done, it is expensive and is better avoided. Extrusions from 0.030-inch to 18 inches outside diameter and 60 inches length can be produced. Labor and tool costs are moderate.

Throwing: A cut blank of plastic material is placed on a revolving plaster disk or table and while turning is shaped by hand roughly with the aid of tools, *Fig.* 5. Hollow blanks with several bore sizes, tapered bores, solid pieces, and round hollow shapes with one closed end are thus produced. External contours and bores are finished ordinarily by turning prior to firing.

Parts up to about 8 inches maximum diameter, 24 inches maximum length or height and ½-inch minimum bore can be made by this method. Output is low, tools are inexpensive but labor cost is high.

Jiggering: With this method, treated material is placed in a revolving plaster-of-Paris mold and worked into the mold contours simultaneously by hand and by means of a profile tool, *Fig.* 6. These partly dried shapes are then trimmed and machined if necessary after further drying. Bowl-shape parts, cylindrical parts with tapered round bores at one or both ends, and bowl shapes with deep skirts can be produced. Sizes ordinarily practicable range up to 22 inches diameter and 10 inches length. Production rate is only moderate, mold and tool expense is moderate, but labor cost is high.

Pressing: In this method a plaster-of-Paris mold is also used, the material being roughly worked to the desired shape. The roughly shaped material, in the mold, is then placed in a suitable press and with an oily, hot, rotating tool or punch is compressed to desired shape, *Fig.* 7. On removal any finishing or machining operations necessary are performed.

Parts similar to those possible with jiggering are produced but ribs and skirts cannot be as deep. Production rate is rapid and ordinarily quantities must be over 1000 pieces. Practicable sizes range up to about 14 inches diameter and 12½ inches length or height. Tooling and setup costs are higher than with the foregoing methods but labor cost is moderate.

Casting: A more recent development of the wet process, casting is probably the most versatile because of the variations in size and shape obtainable. Casting is generally limited to production of parts not possible by other means—irregular shapes not having an axis of rotation, and large parts—although hollow products of revolution are easily made by this method, *Fig.* 8.

Plaster-of-Paris molds are used, the porous mold absorbing the moisture from the mix or "slip" and creating a deposited shell of ceramic. Where inside dimensions are of no consequence, no core is used and the ceramic is merely allowed to deposit to the desired wall thickness before draining the mold. To hold specific internal dimensions and avoid boring, cores are employed to permit build-up of ceramic to the desired shape and wall thickness.

Generally, parts up to 80 inches height and 36 inches diameter are readily obtainable although sections up to 45 inches diameter have been produced. As a rule, weight is limited to about 1400 pounds, maximum. Labor cost is high, a large number of molds is required, and production rate is slow.

Vacuum Extrusion and Pressing: To obtain closer

Fig. 4—Extruded hollow blanks and outline showing design machined before firing

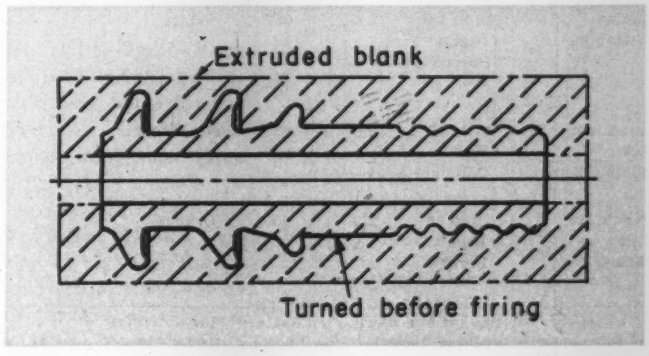
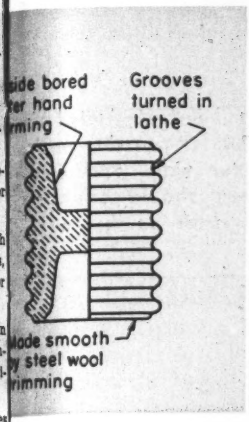


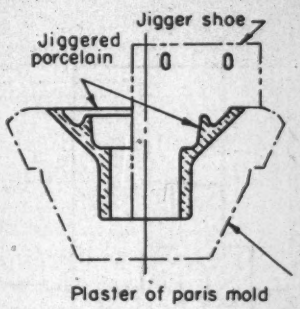
Table 1—Properties of Some Ceramics†

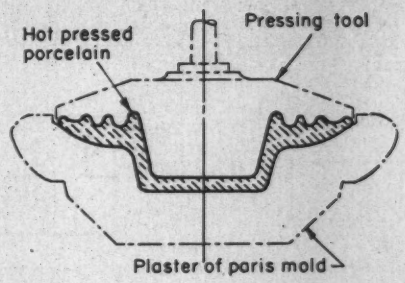
Material	Water Absorption (%)	Spec. Gr.	Density (lb per cu in.)	Standard Colors	Alternative Colors	Safe Temp. (F)	Hardness (Moh)	Tensile Strength (psi)	Comp. Strength (psi)	Flexural Strength (psi)	Properties
Vitreous Ceramics											
Commercial Porcelain§	.5 to 2			White	Green, Brown, etc.	2300		3000 to 3500	30,000 to 40,000		Good quality and most economical type of porcelain for use in most applications.
Steatite AlSiMag 35	0 to .02 impervious	2.5	.090	White	206 Gray 207 Brown	1832	7.5	8500	75,000	18,000	High-strength insulators with good electrical characteristics made to close tolerances for appliances, etc.
Steatite AlSiMag 196	0 to .02 impervious	2.6	.094	White		1832	7.5	10,000	85,000	20,000	Lower dielectric losses than AlSiMag 35. High-strength insulators made to accurate dimensions for r-f circuits.
Steatite AlSiMag 228	0 to .02 impervious	2.7	.098	White		1832	7.5	10,000	85,000	20,000	Excellent mechanical properties with lower electrical losses than AlSiMag 196. For use in high frequency equipment.
Steatite AlSiMag 197	.02 to 1 Vitrified to Tech. Vitreous	2.6	.094	White	209 Gray 210 Brown	1832	7.5	8500	75,000	20,000	Mechanically strong with high electrical resistance at elevated temperatures. For heater wire supports in appliances.
Forsterite AlSiMag 243	0 to .02 impervious	2.8	.101	Buff		1832	7.5	10,000	85,000	20,000	For service where very low loss insulators are required. For ceramic to metal seals. Close tolerances by grinding.
Titanium Dioxide AlSiMag 192	0 to .02 impervious	4.0	.144	Tan		1832	8	7500	80,000	20,000	Ceramic for polished thread guides. AlSiMag 193 is the same mechanically but made conductor to dissipate static.
Zircon AlSiMag 475	0 to .02 impervious	3.7	.134	White		2012	8	12,000	90,000	22,000	Vitrified ceramic with excellent electrical properties, high strength and good resistance to thermal shock.
Alumina AlSiMag 491‡	0 to .02 impervious	3.5	.125	Blue		2552	9		100,000	45,000	Fine texture, extremely hard, resistant to abrasion, chipping and corrosion. Best adapted to small machined or pressed parts.
Alumina AlSiMag 513‡	0 to .02 impervious	3.4	.122	Gray		2462	9		100,000	45,000	High-strength alumina ceramic for somewhat larger size articles than possible in 491 by machining or pressing.
Lavas											
Aluminum Silicate Grade A Lava	2 to 3 Technically Vitreous	2.3	.085	Pink		2012	6	2500	20,000	9000	Good electrical and heat resistance, excellent for close tolerances. Available in larger sizes than other lavas.
Magnesium Silicate Lava 1136	2 to 3 Technically Vitreous	2.8	.102	Tan	White	2282	6		25,000	9000	Excellent electrical properties, made to close tolerances. For insulating spacers in small size vacuum tubes.
Refractories											
Cordierite AlSiMag 202	*	2.1	.076	*	*	2282	7	3500	40,000	8000	Low coefficient of expansion, high resistance to heat shock. For heating elements, thermocouple insulators, burner tips.
Zirconium Oxide AlSiMag 550‡	6 to 10 Semi-vitreous	3.8	.136	Orange		2912			30,000	10,000	Fine grain, high mechanical strength refractory for simple extruded shapes. Has excellent resistance to heat shock.
Zircon AlSiMag 504	8 to 14 Highly Porous	2.9	.105	White		2462			16,000	13,000	Very fine grain refractory with good resistance to heat shock. Suitable for firing plates and supports, brazing jigs, etc.
Alumina AlSiMag 393	12 to 18 Highly Porous	2.4	.087	White		2552			11,000	10,000	Easily degassed refractory with good electrical resistance at high temperature. Porous body principally used for vacuum tube insulators.
Magnesium Silicate AlSiMag 222‡	14 to 18 Highly Porous	2.0	.072	Light Brown		2372	6	2500	25,000	8000	Refractory which is machinable in the fired state. Sold in rods, tubes, plates and blocks. For models or repair parts.
Zirconium Oxide AlSiMag 508‡	12 to 16 Highly Porous	2.9	.106	Orange		2912			11,000	4000	Refractory body excellent for heat shock. Made only in plates, rods and simple shapes by pressing or casting.

† Courtesy American Lava Corp. § Courtesy Star Porcelain Co. ‡ Economic limitations on sizes and shapes.

* Furnished in various densities: 202, porous, 10 to 15% (tan), 178, porous, 10 to 15% (white), 547, semi-vitreous, 2 to 7% (brown)







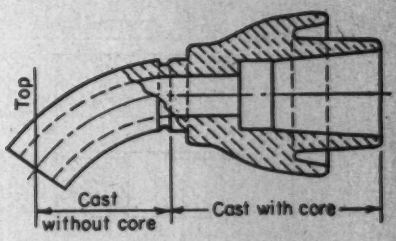


Fig. 5—Left, above—A typical hand thrown and turned mechanical procelain part

Fig. 6—Left, center—A porcelain fitting which is jiggered in a plaster mold and trimmed or turned

Fig. 7—Right, center—A typical porcelain part hot pressed with a rotating tool in a plaster mold

Fig. 8—Extreme right, above—Special shape cast partly with a core and partly without a core where it is impractical

Fig. 9—Right—Complicated part pressed in evacuated dies. Charge is weighed to insure accuracy

dimensional control and impervious structure, ceramic with reduced water content is processed specially by evacuation of the mix before forming in steel dies. Machining is ordinarily not needed.

Best suited for simple shapes and pieces, this method is economical where large quantities are the order—say 5000 pieces or more. Extrusion is limited to outside diameters up to 1.625 inches, lengths to 10 inches and inside diameters up to 0.140-inch. Pressing, *Fig.* 9, is limited to a maximum height of 7 inches, maximum length of 12 inches, maximum width of 10 inches, and projected areas up to 120 square inches. Tool and setup costs are high and labor cost is moderate.

DRY PROCESS: Differing from the plastic process of forming in that only a semimoist granular powder is used, the dry process is subject to much less dimensional variation in manufacture. Pressing in metal dies is the only method of forming used. Typical parts possible are: Flat or ribbed plates of any form or curvature with or without holes, slots, recesses, and barriers; U and other shapes; intricate box shapes with ribbed partitions and side holes or slots. Machining or other finishing is seldom required, *Fig.* 10.

An economical method for producing large quantities at moderate cost, dry pressing is generally limited to relatively small parts in the neighborhood of 2 to 4 inches. Maximum length or width possible is 14 inches, height is generally limited to about 7 inches and projected area to 196 square inches.

High-clay ceramics generally result in a porous structure with this method but the low and clay-free types can be produced with excellent results. The close dimensional control possible makes the process advantageous with the latter groups. Dry pressing is

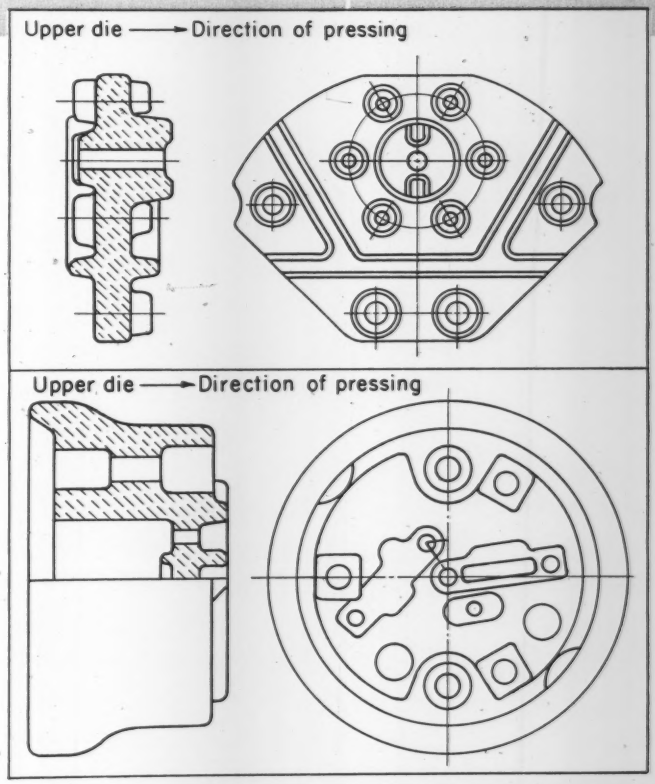


Fig. 10—Above — Porcelain electrical part designed for dry pressing in metal dies

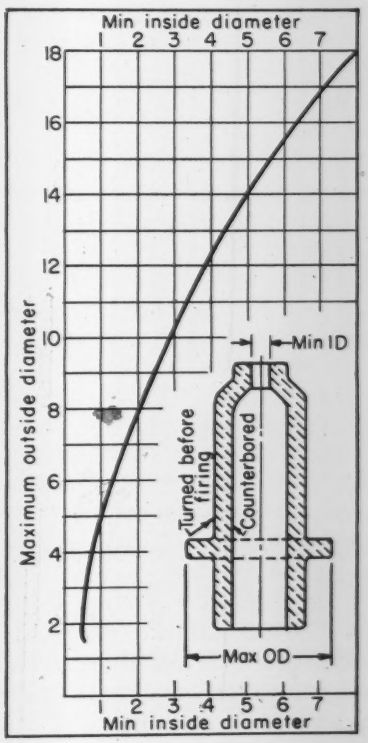


Fig. 11—Right—View of part machined inside and out after extrusion. Chart indicates desirable proportion of inside to outside diameters for extrusion. Holes as small as 0.010-inch and outside diameters of 0.030-inch have been produced

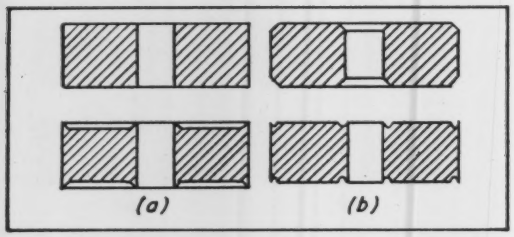
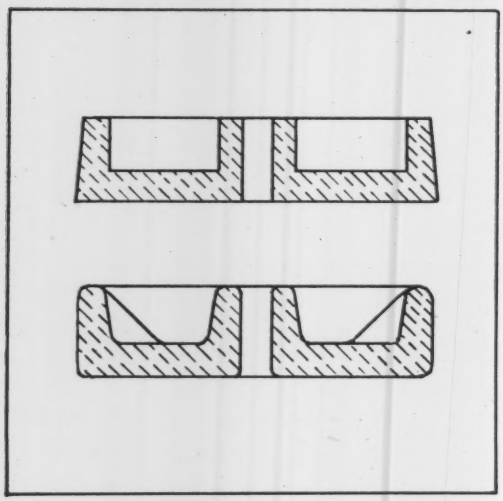


Fig. 12—Above—Sharp edges on pressed parts (a) are less desirable than bevel edges (b) owing to better appearance and less fin trouble after firing

Fig. 13—Below—Dry pressed part showing improved design with strengthening rib



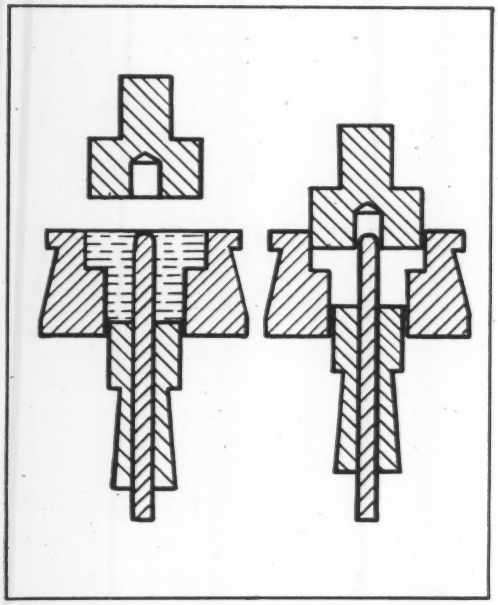
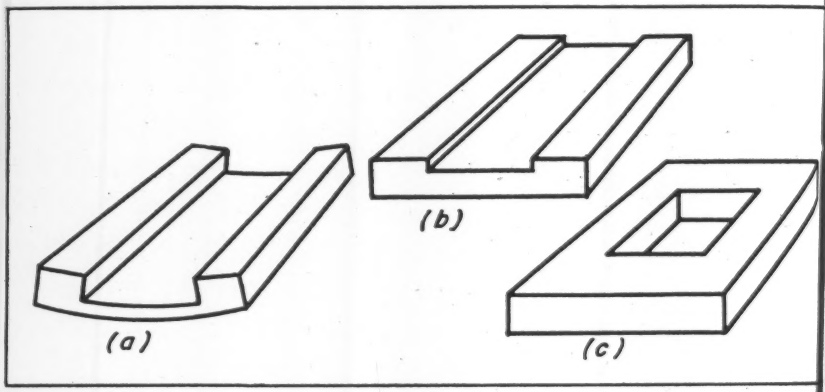


Fig. 14—Left—Metal die setup for pressing parts. Outer walls and core piece should have no draft as shown

Fig. 15—Below — Deep groove in part (a) results in warpage. Shallow groove design (b) is better but a recessed design (c) is most desirable alternate



well adapted to high production, and automatic presses may be used.

DESIGN CONSIDERATIONS: Structurally, the problem of designing with ceramics resolves itself into one of design to compensate for or avoid involving the undesirable lack of ductility and low tensile strength. The best design procedure is to take all possible advantage of the high compressive strength of these materials and avoid as much as possible all loading under tension. Pronounced weakness at notches makes it imperative to avoid sharp change in section thickness or direction.

Wall Thickness: Where hollow extrusions are designed the wall thickness may range from 10 to 20 per cent of the outside diameter, but should be held to 1¾ inches, maximum, if possible. Rectangular or square shapes require liberal internal fillets of at least ½-inch radius. The cross section of such parts should be held as uniform as possible to aid drying. Thus, concentric external radii should be used to match the internal fillets, and wall thickness should be held to 1¼ inches or less.

Where heavy walls are necessary for external machining on hollow extruded sections, wall thickness should be minimized to avoid high losses due to differential in shrinkage of finished items, *Fig.* 11. Sections in any case should be kept as uniform as possible without abrupt variations. As a general rule, sections for wet pressing, should not exceed 1 inch.

Fillets and Radii: Along with uniform section thickness, generous fillets should be employed to reduce stresses and aid the flow of material in forming. Fillets at bosses, shoulders, webs, ribs, depressions, or counterbores should depend upon the height of these features. For those features formed by the mold or die, the minimum radius should be equal to the depth or height of the adjoining feature for dimensions of ¼-inch or less. With sizes from ¼-inch to 2 inches, the minimum radius should be ¼-inch and for those over 2 inches the radius should be at least one-eighth the depth or height. For parts to be machined, smaller fillet radii are practicable. On dimensions up to 3/32-inch the radius can be equal to the height or depth, from 3/32 to ¾-inch

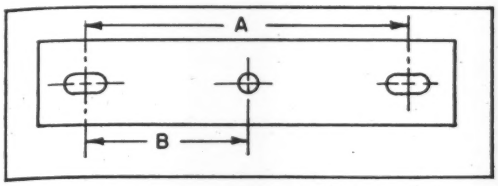
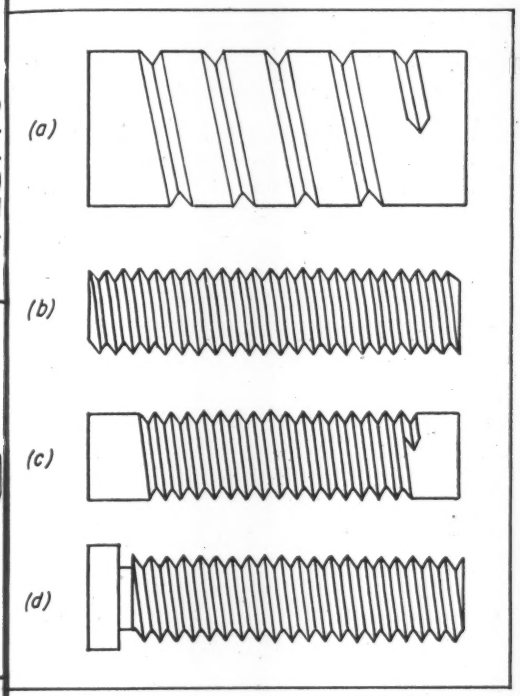
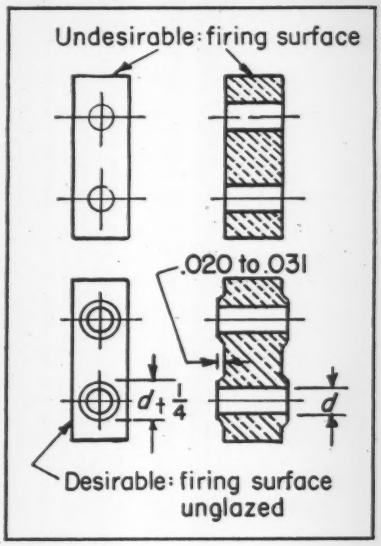


Fig. 16—Above—Holes should be elongated purposely to overcome the effect of firing shrinkage. One hole elongated requires twice the amount for two





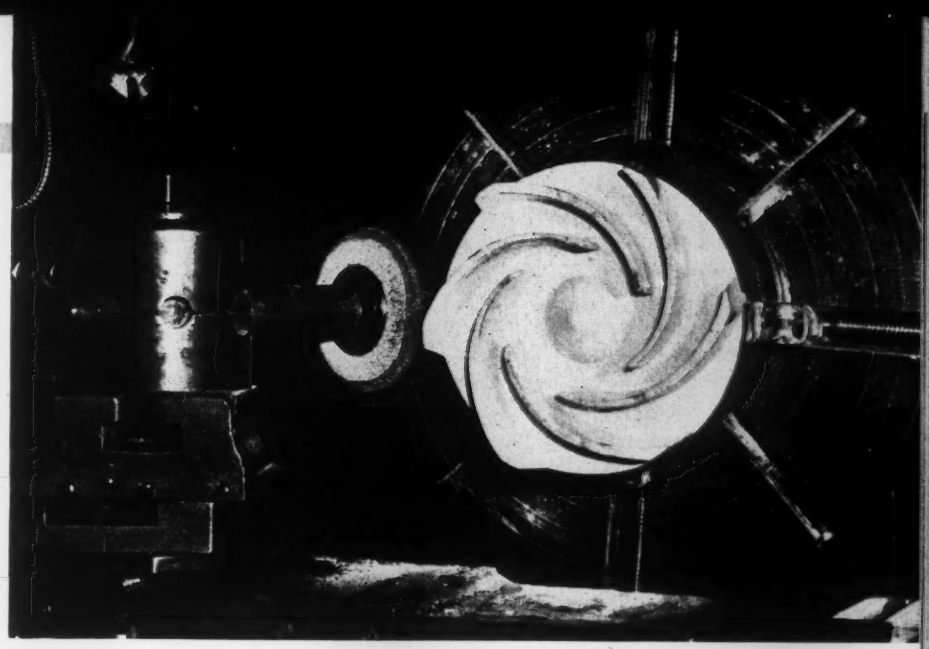


Fig. 17 — Extreme left — Extended-pitch threads (a) are preferable. Full length (b) is more economical than (c) or (d)

Fig. 18—Left—Where ground surfaces are required for fit to metal parts, reduced boss area is preferred design

Fig. 19—Above—Centrifugal pump impeller being finish ground to close machine tolerances

values the minimum radius should be 3/32-inch, and for deeper or higher features it should again equal at least one-eighth the dimension.

Corners or edges should be rounded off to improve appearance and obviate chipping. Minimum radius practicable is 1/16-inch and larger values are preferable. Beveled or chamfered edges are generally desirable on dry pressed parts. A standard chamfer is generally accepted at 1/32-inch by 45 degrees. The bevel not only reduces die wear but simplifies fin removal after firing, *Fig.* 12.

Ribs: Protruding ribs or fins should be spaced apart at least as much as they are wide or deep. A full radius on the top of ribs is advantageous. Where walls are relatively light, ribs should be employed for strengthening and to prevent distortion, *Fig.* 13.

Draft: Some draft is required on parts for most processes used. In forming, both molds and cores must be withdrawn and at least 1/64-inch per inch should be provided although 1/32-inch per inch per side is more desirable. Large cores should have not less than 2 degrees per side. For dry pressing, plain outside surfaces and through holes should be made without draft, *Fig.* 14.

Grooves: Deep grooves which run over the length of a part tend to cause warping. To avoid this condition grooves should be kept shallow and not over one-half the piece thickness if possible. A recess is preferable wherever practicable in lieu of a channel shape, *Fig.* 15.

Holes: In general, holes should be 1/16-inch diameter or over for economy, although holes as small as 0.010-inch are produced in some parts. Wall thickness between holes should be as substantial as possible. The same is true of walls between holes and edges. Thin walls in dies are fragile and may be expensive. Wherever practicable, walls between holes should equal at least one-half the hole diameter and should never be less than 1/32-inch in any case. Thinner walls are sometimes possible but it is inadvisable to have walls under one-third the hole diameter. Walls between holes and edges should never be less than ⅛-inch.

The firing shrinkage along the axis of extrusion or

in the direction of pressing is different from that perpendicular to the axis or direction of pressing. Holes, therefore, tend to become somewhat elliptical, as much as 2 to 3 per cent being common. The minimum diameter of clearance hole to permit assembly of bolts, pins, studs, etc., can be determined from:

Diam. of Fastener (in., max.)	**Clearance** (in., min.)
1/16 to 5/32	0.006
3/16 to 3/8	0.008
13/32 to 5/8	0.010
21/32 to 1	0.012
over 1	0.016

Maximum diameter of hole can then be found by applying the expected total tolerance variation, plus.

Shrinkage also affects hole spacings, and generally it is undesirable to attempt to have round holes match the hole spacings in a metal attachment. One or all of the holes in the ceramic part should be elongated to provide ready assembly. Two holes should be elongated in the direction of maximum shrink by an amount equal to the total tolerance on the distance between hole centers, *Fig.* 16. If only one of two holes is to be elongated, the amount of elongation should be equal to twice the tolerance on the distance between hole centers.

Threads: It is impractical to mold threaded holes in ceramics. Coarse tapped threads are preferable and resist crumbling found with fine pitches. Smallest tapped hole which can be made satisfactorily is a number 3-48. Smaller sizes are not recommended. The number of threads in contact should be kept at the minimum owing to shrinkage. It is seldom necessary to use over 12 threads engagement and the ideal fit is obtained with about eight threads engagement in a 12-thread hole. For shallow holes, a screw two-thirds to three-quarters of the tapped depth is adequate. Where screws must pass through a part, the thickness and pitch should be selected so that six or seven threads are used, the minimum being four and the maximum, eight. Holes should be counterbored before tapping to reduce losses from cracking.

External threads may be cut sharp but are more durable if an extended pitch can be used, *Fig.* 17*a*. Fillets and radii for threads, as with grooves and slots, are recommended to add to the strength and to resist corner breakage. Threading is most economical if full length, *Fig.* 17*b*. A portion is more expensive, *Fig.* 17*c* and to a shoulder or undercut most expensive, *Fig.* 17*d*. Contact on external threads should be from six to twelve threads. Metal parts should be proportioned to keep contact under twelve.

Machining: Parts which must be otherwise machined before firing require some consideration. External fillets should have at least 1/8-inch radii if possible. Deep undercuts should be avoided. Since internal machining is extremely expensive it should be avoided. In turning external shapes the final piece must be capable of resisting the loads without breaking. With solid bodies, the ratio between the length and root or minimum turned diameter should not be greater than 6 or 7 to 1.

Where parts must fit flat surfaces after firing, grinding is necessary. Maximum economy is possible by employing a group of bosses to reduce the

Fig. 20—Group of mechanical parts including special textile machine thread guides and other wear-resistant inserts

area to be finished. Bosses around holes should be about ¼-inch greater in diameter than the hole. Boss height may be about 0.020-inch but 1/32-inch is preferable, *Fig.* 18.

Almost any required accuracy can be had by grinding but stock removal is expensive. However, numerous grinding operations are regularly employed in mechanical ceramic and stoneware parts where corrosion resistance, hardness, etc., are imperative, *Fig.* 10.

Firing Surface: To permit solid resting of parts during firing in the kiln, a firing surface should be left unglazed, *Fig.* 18. Long flat pieces should be fired on one long side to reduce warping.

Ordinarily tubular and bushing parts with a height not exceeding six times the diameter of the base are sufficiently steady for foot firing but longer parts must be hung. Hanging is expensive and generally pieces with up to 160 cubic inches of volume per square inch of supporting area can be fired in this manner. Designs longer than fifty or sixty inches should be avoided.

Lettering: Block lettering with raised characters on a depressed surface is preferred for dry pressed parts. All others are generally marked with ceramic ink on unglazed portions.

MATERIALS: Ceramic materials are generally chosen for mechanical or industrial equipment because of corrosion, heat or wear resistance, *Fig.* 20, or for electrical properties. Design to obtain the advantage of these properties must take into consideration the relatively low strength and brittle character of these materials. In TABLE I the general physical properties of some ceramics are listed.

PRODUCTION PROCESSES IN BOOK FORM

Design of parts for economical manufacture is covered in the form of practical and down-to-earth data on some 36 separate processes in the new book, *Production Processes—Their Influence on Design,* based on this series of the same name. Containing 568 pages and 652 illustrations, the 6½ by 10-inch volume, clothbound, is available from MACHINE DESIGN at $10.00 postpaid.

TOLERANCES: With porcelain parts the combined drying and firing shrinkage runs from one-seventh to one-eighth, kiln shrinkage alone being one-tenth to one-twelfth. With such conditions the problem of tolerances is critical, especially since shrinkage is not always uniform. It has been established that a standard tolerance between glazed and unglazed surfaces of plus or minus 3 per cent or 0.03-inch per inch or fraction thereof will permit practical commercial manufacture with reasonable cost for wet process porcelain. With dry process porcelains, the practicable limits are somewhat less than those for the wet process—about 2 per cent, or plus or minus 0.02-inch per inch. Minimum single tolerance on dimensions less than 1 inch with the former is 0.03-inch, plus or minus, and with the latter is 0.02-inch.

Where close or so-called critical tolerances must be specified, cost is increased but generally a tolerance of 2 per cent plus or minus with a minimum of 0.020-inch in the case of wet process parts and 1½ per cent, plus or minus, with a minimum of 0.015-inch for dry process parts can be obtained. Dimensions between unglazed surfaces of parts can generally be held to about one-half these figures if necessary.

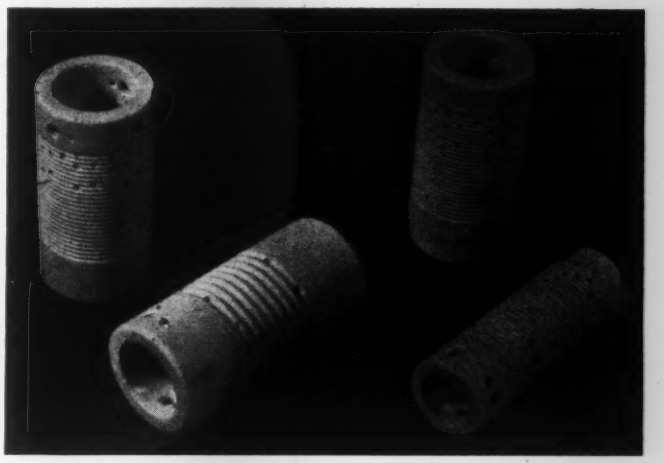


Fig. 21—Group of electronic coil forms showing numerous plain and tapped holes

For dry process pressed parts, dimensions determined primarily by the upper and lower die positions or those affected by side wear are subject to the pressures used and variations in amount of fill. Such dimensions should have tolerances increased by 50 to 100 per cent, if possible.

Standard practice with dry process low-clay technical ceramics such as steatite is to hold plus or minus 1 per cent on critical dimensions over unglazed surfaces with a minimum of plus or minus 0.005-inch on general dimensions and plus or minus 0.010-inch on thickness dimensions. Over glazed surfaces the limits are plus or minus one per cent plus 0.010-inch, with a minimum of plus or minus 0.015-inch. Where thickness is in excess of one inch, the recommended tolerance is plus or minus 3 per cent. All noncritical dimensions should carry the standard critical plus 0.015-inch.

On hole spacings, a tolerance of plus or minus 0.005-inch between centers measuring under ½-inch and plus or minus one per cent on larger dimensions can be held. On unglazed holes standard tolerances are plus or minus 0.005-inch for holes up to ½-inch and plus or minus one per cent for ½-inch and larger holes. For glazed holes these tolerances are increased to plus or minus 0.012-inch for diameters up to 0.600-inch and plus or minus 2 per cent on holes 0.600-inch and greater. Screw threads, *Fig.* 21, conform generally to American Standard Class 1 fits.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

American Lava Corp. (Figs. 1, 12, 14, 15, 16, 17 and 20) ..... Chattanooga, Tenn.
Henry L. Crowley & Co. (Fig. 21) ..... West Orange, N. J.
Locke, Inc. (Figs. 2 and 3) ..... Baltimore, Md.
Star Porcelain Co. ..... Trenton, N. J.
U. S. Stoneware Co. (Fig. 19) ..... Akron, Ohio
Westinghouse Electric Corp. (Figs. 4, 5, 6, 7, 8, 9, 10, 11, 13, and 18) ..... Pittsburgh, Pa.

Designing Low-Cost Forgings

By J. J. Sloan and K. R. Denny
Production Design Engineers
North American Aviation Inc.
Los Angeles, Calif.

GIVEN equal costs, die forgings are generally preferred to castings or extrusions because of their high strength-to-weight ratio. The nature of the forging process is such that it shapes the grain of the metal into approximately the direction required by design, a feature which usually cannot be obtained by other methods unless it so happens that the loads are parallel to the grain in a bar or extrusion. Good design of a forging is essential and through good design, forgings can be made comparatively cheaply with excellent functional properties. The total cost of a forging must include its prorated tool cost, price per piece as supplied by the forging vendor, and the labor and machine time expended on completing the piece.

In selecting the material for die forging the accompanying table of strength-to-weight ratios is useful, based on the tensile yield strength of 14S-T aluminum alloy as 100 per cent. In the operation of closed dies the parting line shown by the drawing is of great importance since excess metal makes its exit at the parting line. *Fig.* 1 shows the effect on grain direction, and consequently strength, of placing the parting line at various positions on a channel section. By placing the parting line as at *a*, a cheaper die can be made; machining of the impression is necessary in only one die block. When placed as at *f*, the parting line is least desirable because of grain direction.

Steel forgings are best designed with impressions in both die halves but with most of the projection into the upper half. Scale will drop out of the upper impression and may be blown away from the work and even a slight impression in the lower half will materially assist in locating the metal for forging.

Die costs are in accordance with the size of the die and the amount of machining necessary to sink. In buying forgings by the piece, the original die cost is submitted as tool cost and must be prorated over the number of pieces purchased. Maintenance of the die is figured into the "per piece" price, since dies sometimes require replacement over long runs. The effect of parting line on die cost is exemplified in *Fig.* 2.

Where left and right-hand parts are similar, consideration should be given to making each part from

Fig. 1—Effect of parting line on grain direction

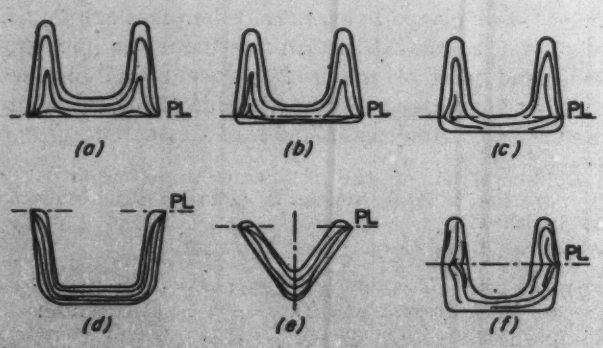
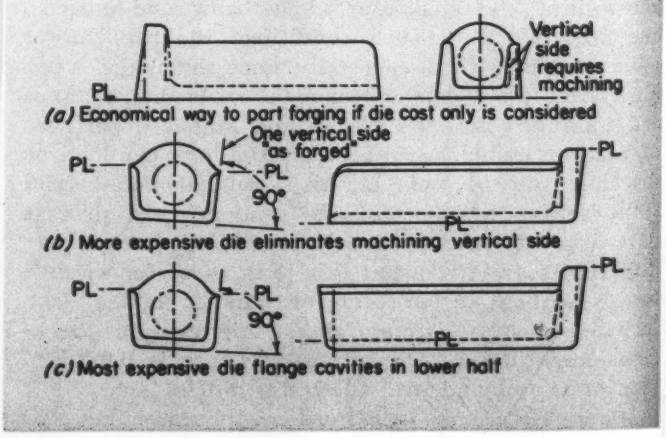


Fig. 2—Methods of placing parting line to reduce die cost



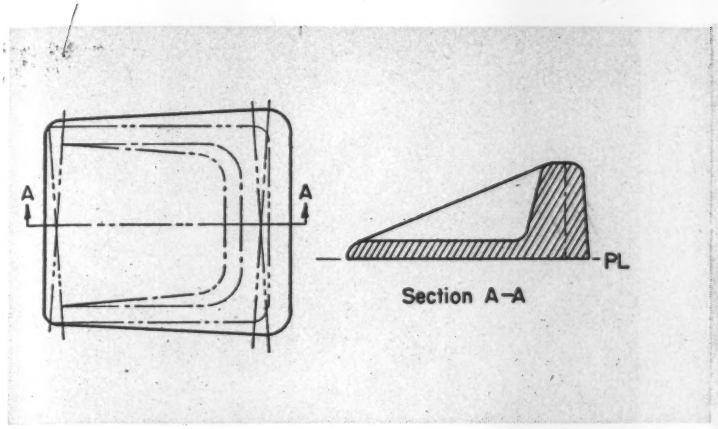


Fig. 3—Above—Neutral forging from which left and right-hand parts can be machined

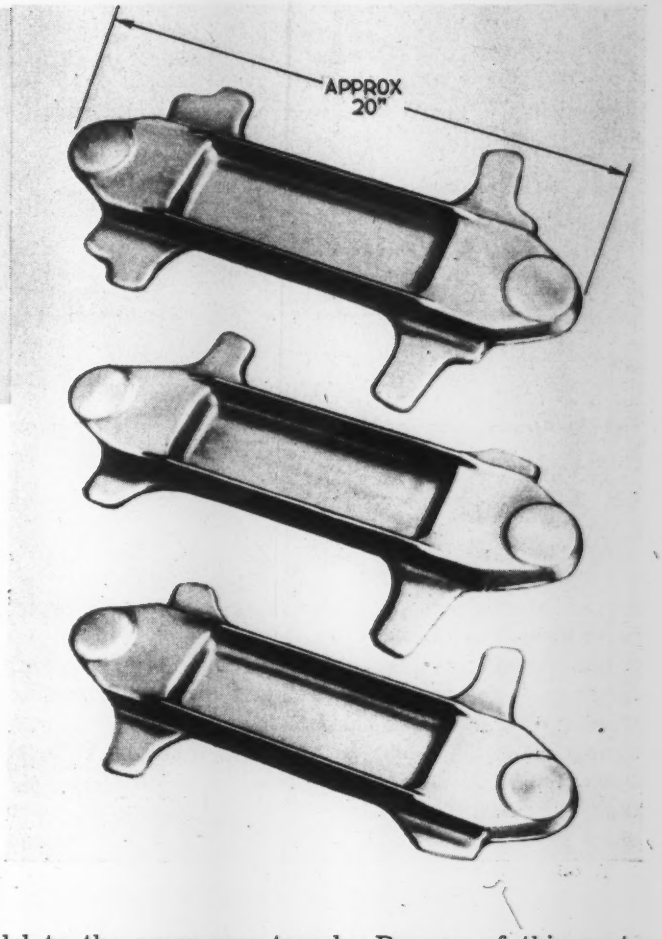


Fig. 4—Right—Left and right-hand parts made by cutting off projections

a neutral forging. This can be done by leaving extra material for face cuts at different angles as in *Fig.* 3 or by forging a profile such that each left or right can be made by lopping off opposite arms, *Fig.* 4. In some cases the number of pieces of each hand is limited but a great many are required altogether.

Forgings are sometimes made with both right and left-hand parts in the same die in order to balance the die, or to obtain better material flow in thin flanges of separate parts, *Fig.* 5. The saw cut and machining which follow later reduce the web thicknesses to required dimensions with better grain direction.

In *Fig.* 6 is shown a forging which includes left and right-hand parts. Either part could be forged with the line bisecting the angle between the flanges parallel to the press ram travel. Because of this acute angle and its affect on the life of the die it is better to forge left and right together with only 14 degrees between two corresponding flanges, thereby increasing the usage of the die.

Draft angle is equal in importance to the parting line since it affects die shape and costs. About 5 to 7 degrees of draft are usually required on all forgings and this plus large fillet radii add weight to the part. If all draft and large fillets must later be entirely removed by machining it may not be economical to die forge. Also, the draft usually occasions spotfacing for bolts or other fasteners.

A few simple sections can be forged with 7 degrees draft and ironed into shape with no draft. Other parts can be adapted so that no draft is necessary by properly placing the parting line. *Fig.* 7 shows various parts positioned in their dies, *a, b, c* and *d* having the required 7-degree draft. *Fig.* 7*e* is a part with backdraft. Actually the 14-degree draft shown permits ejection of the part from the die and may be removed by machining. Dies of this nature are subject to approval by the vendor. In *Fig.* 7*f* is shown a locked die, so-called because upper and lower die halves are mated together in vertical alignment by their tongue and groove shape. This is a more expensive type of die but must be balanced against the machining factors of the part and in some cases proves least expensive. The increase in cost for dies which forge without draft, or with back-draft, may be from 20 to 50 per cent. Construction of locked dies may increase the cost 100 per cent over other types. Piercing of holes under 2 inches diameter in

Comparison of Strength-to-Weight Ratios

Material	Approx. Ratio (%)	Application
14S-T Al. Alloy Spec. QQ-A-367 Class 5	100	Standard high-strength forgeable alloy. Not welded.
75S-T Al. Alloy* Spec. AMS4139	128	Not as easily forged as 14S-T. Highest strength. Not welded.
0-1 Aged Magnesium Alloy Spec. AN-M-21	64	Where rigidity by mass is required. Because of susceptibility to corrosion requires careful consideration. Not welded.
Hardened & Tempered 4130, 4140 and 4340 Alloy Steel	60 to 125	Used where temperatures are greater than allowable for aluminum alloy or where space is limited or bearing loads require high strength. Weldable.
1020 Carbon Steel Normalized	26	Used where case hardening is necessary for wear and for small lightly stressed parts. Strength/weight ratio based on normalized forgings. Weldable.
Corr. Res. Steels Types 321 & 347 & Super Alloy Steels Annealed	21	Used for corrosion resistance and strength at elevated temperatures. Weldable.
396 & 195 Aluminum Casting Alloys	42	Used for intricate fittings where forging strength is not required. 195 alloy is weldable.

* The use of 75S alloy is limited to designs approved by the chief project engineer. Factors present in the forging of any aluminum alloy are more critical with 75S and require closer control.

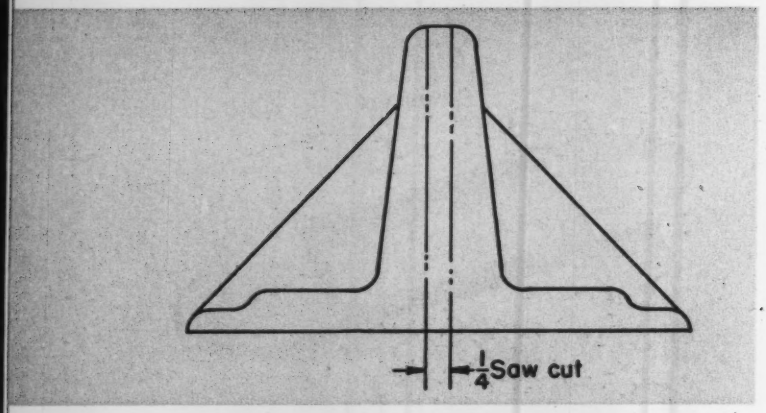


Fig. 5—Above—Siamese forging to obtain better material flow in thin flanges

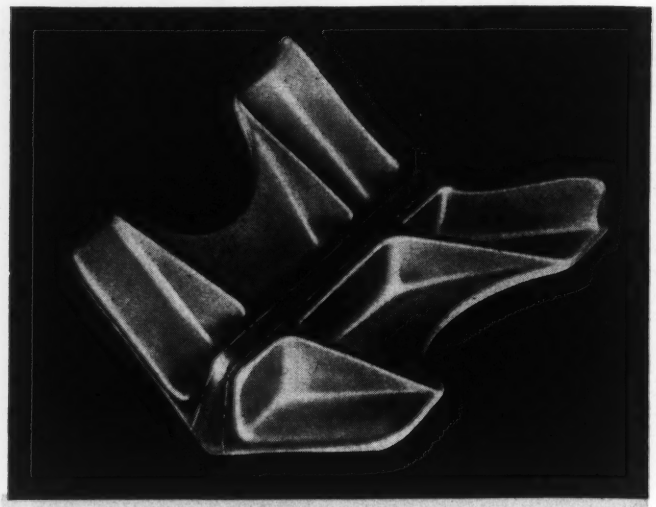


Fig. 6—Right—Left and right-hand parts forged together to eliminate backdraft

forged aluminum alloys is not economical and such holes should be drilled.

Some parts are adaptable as extrusions rather than forgings. The important point is die cost, since extrusion dies are much cheaper than forging dies. Here again, machining cost on the part must be weighed but extrusions are generally much cheaper than forgings in small quantities. *Fig.* 8 indicates various shapes which may be better cut from extrusions. *Fig.* 8*a* had much less finish machining cost since the part was extruded without the draft on top and bottom sides of the flanges.

Some design considerations for die forgings are:

1. Where grain direction is critical it must be specified on the drawing
2. Place the parting line with regard to strength and to minimize machine time both on die and part
3. Draft of 7 degrees is required, 5 degrees and backdrafts are subject to vendor approval
4. Minimum web thicknesses for 14S alloy in general design is:

Area (sq in.)*	Minimum Web (in.)
Up to 4	3/32
4 to 8	7/64
8 to 20	1/8
20 to 40	5/32
40 to 80	3/16
80 to 150	1/4
150 to 250	5/16
250 to 500	3/8

* Area in view perpendicular to forging direction. For 75S alloy use next greater corresponding thickness.

5. For corner radii, fillet radii and tolerances on die forgings see Drop Forging Association Standards.

Although most aircraft forgings are produced on presses of 1500 tons or less, or on hammers of 2000 to 12,000 pounds capacity, it should be noted that larger equipment is available and being designed. The largest to date in the United States is the 18,000-ton forging press at Grafton, Mass., housed by Wyman-Gordon Products. Designed for light alloys, magnesium and aluminum, this press, with a platen 7 by 12 feet, is capable of making very large forgings providing the opportunity to simplify and improve structures to an extent heretofore impractical. It should be noted that dies are very expensive in such large sizes and subsequent engineering changes may be extremely costly to carry out.

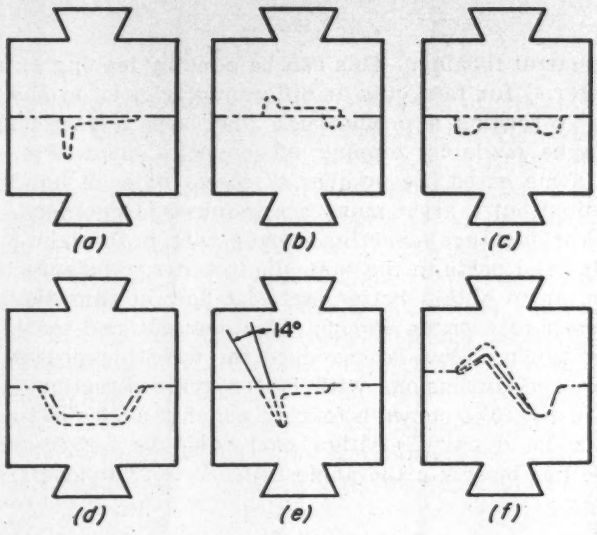
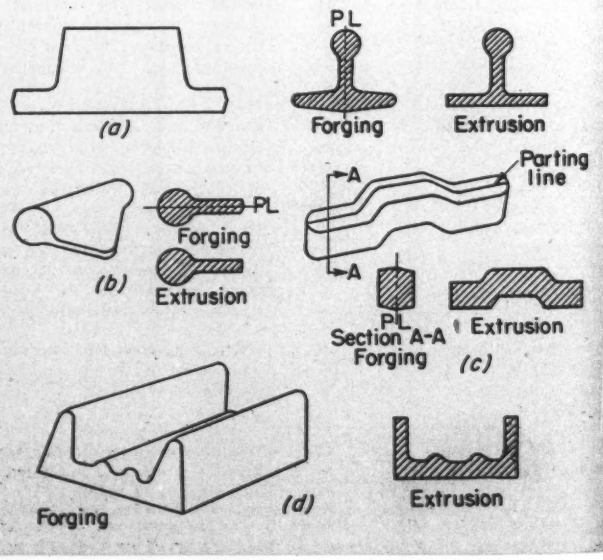


Fig. 7—Above—The effect of draft angle on die design

Fig. 8—Below—Forged shapes which should be less expensive as extrusions



Why Machine Parts Fail

Part 4–Fatigue Fractures

By Charles Lipson
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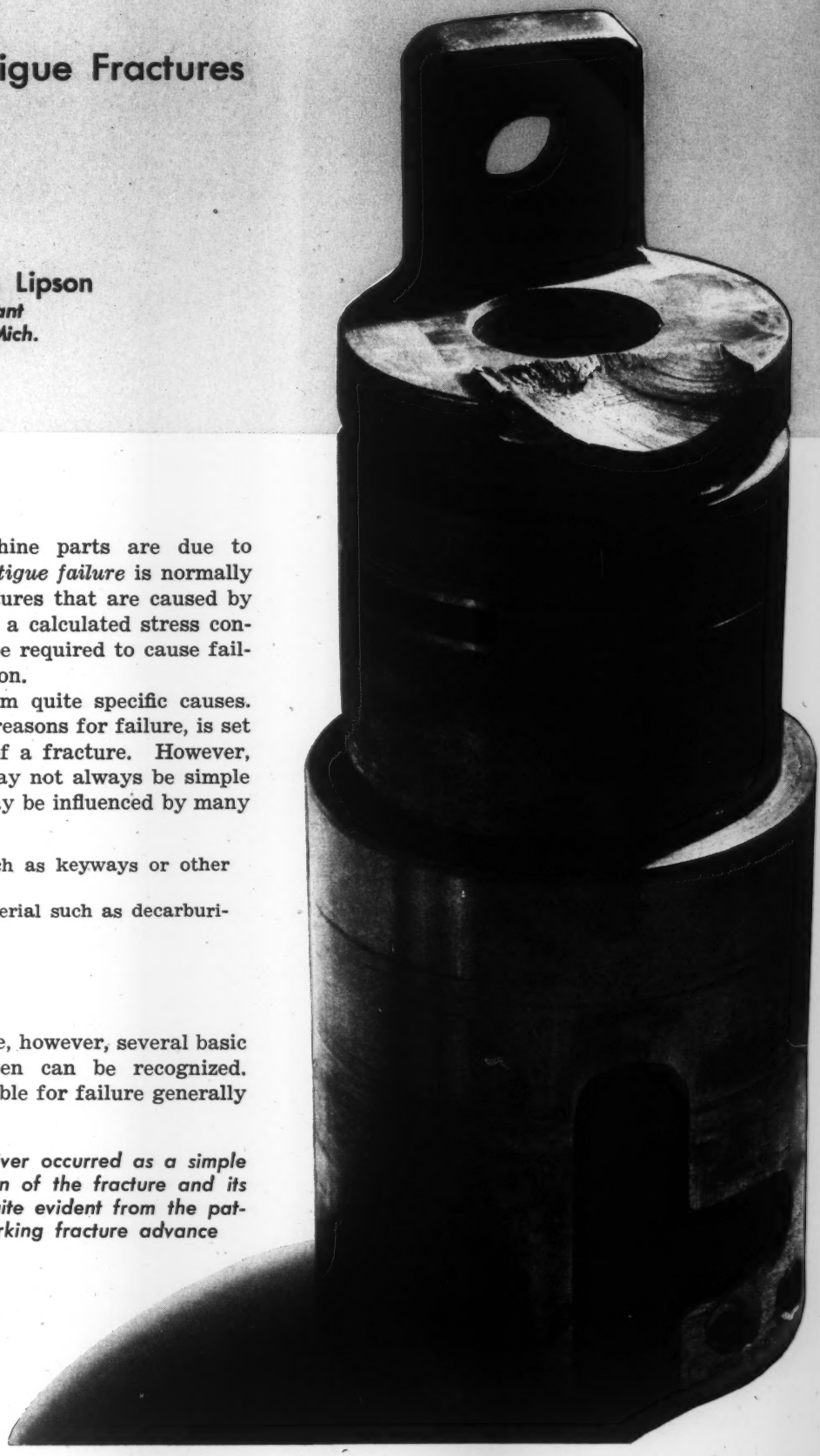
Detroit, Mich.



MOST failures in machine parts are due to fatigue. The term *fatigue failure* is normally applied to those fractures that are caused by repeated load applications at a calculated stress considerably lower than would be required to cause failure in a single load application.

Fatigue failures result from quite specific causes. The complete story, with all reasons for failure, is set forth in detail on the face of a fracture. However, interpretation of fractures may not always be simple or easy, because each case may be influenced by many variables such as:

1. Adverse form of part, such as keyways or other severe contour changes.
2. Lack of uniformity of material such as decarburization of surface.
3. Residual stresses.
4. Surface conditions.
5. Resonant vibrations.

BASIC PATTERNS: There are, however, several basic fracture patterns which often can be recognized. From them the loads responsible for failure generally

Fig. 36—Failure of this hone driver occurred as a simple fatigue fracture. Point of origin of the fracture and its direction of propagation are quite evident from the pattern of stop marks, lines marking fracture advance

Photo, courtesy Micromatic Hone Corp.

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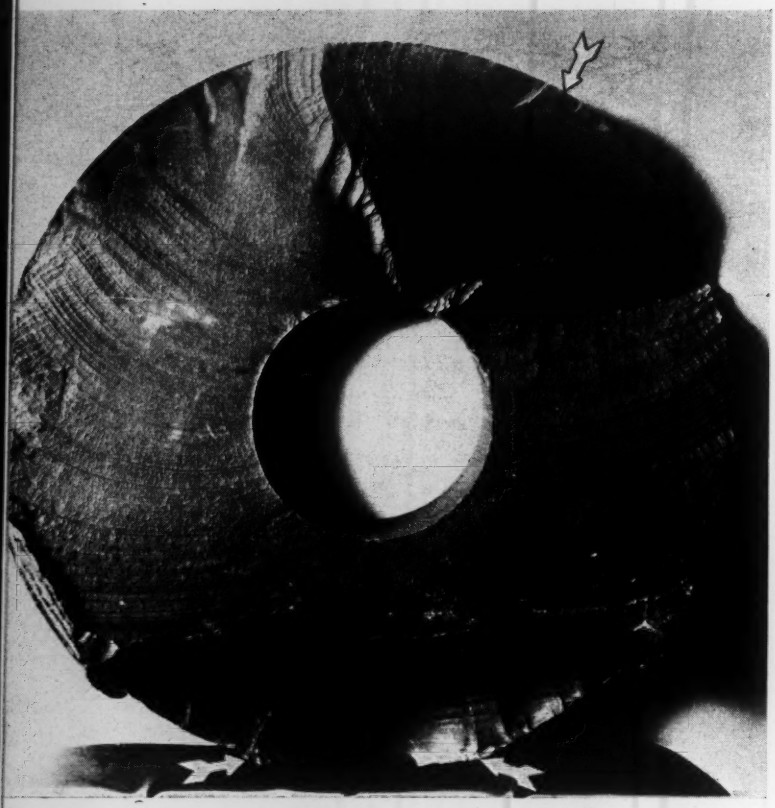
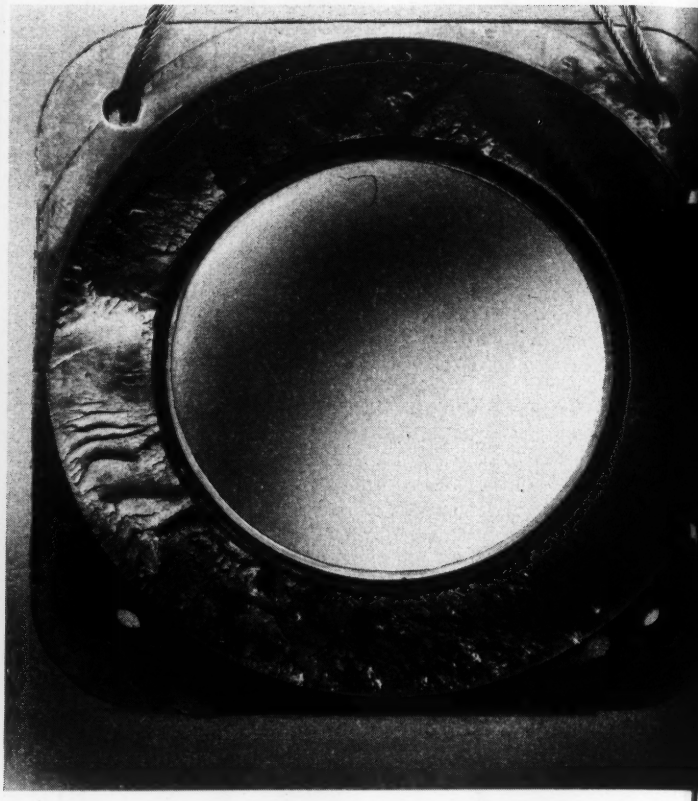


Fig. 37—Originating at three separate points marked by arrows, failure of this locomotive main driving axle occurred as a compound fatigue fracture. The cracks spread until joined in final rupture



Photo, courtesy Briggs Manufacturing Co.

Fig. 38—This compound fatigue fracture of a hydraulic cylinder resulted from nonuniform stiffness in the mounts and lack of fillets. Two cracks originated at the points marked by arrows

can be determined. Where complex loading may have existed prior to failure, a peculiarly shaped fracture possessing characteristics of more than one basic pattern can be expected. Fundamentally, however, the origin of failure can be found by locating the center of radiation of the fatigue waves. One must look not only to the origin of failure but must complete the picture by determining both the type of load which has been acting and the existence of any possible additive factor, either initially present or developed after the first formation of a fatigue crack. An example of a developed additive factor is resonant vibration which may be set up in a member subjected to torsion as the result of change in characteristics after a crack has started.

The point of origin of a failure can be traced to any of a number of causes: surface imperfections, grinding marks, forging marks, and stress raisers such as keyways and fillets. It is often observed that there are several points of origin. In such a case the resultant fracture pattern is affected by the influence of each initial crack whose consequent wave form will act jointly with the wave forms of the other initial cracks.

TWO KINDS OF FRACTURE: Fractures in general can be described as *simple* and *compound*. A simple fracture results from a single crack that spreads across the cross section and causes ultimate failure. A compound fracture results from two or more cracks that spread from different points on the periphery of the section and cause a joint effect on the fracture appearance. This effect is apt to be particularly noticeable in fractures of highly overstressed splined shafts.

The simple fracture shown in *Fig.* 36 implies the initial existence of a point on or near the surface that was weaker than any other on the periphery of that particular transverse plane. This point, therefore, became the sole origin of the crack. It may be thought that the weakening effect of this first crack would cause cracks to start forming at other weak sections, but this "touching off" action is rarely shown in experiments. Each of the compound fractures shown in *Figs.* 37 and 38 seems to have been produced by two or more individual cracks originating in different highly stressed points.

TWO DISTINCT ZONES: Fatigue fractures occur without perceptible ductility. The actual fracture face is composed of two distinct regions: one smooth and velvety—the *fatigue zone,* and the other coarse and crystalline—the *instantaneous zone.* The nucleus of a simple fracture will usually be evident at some point on the fracture face. From this origin, after many cycles of stress, the crack propagates itself

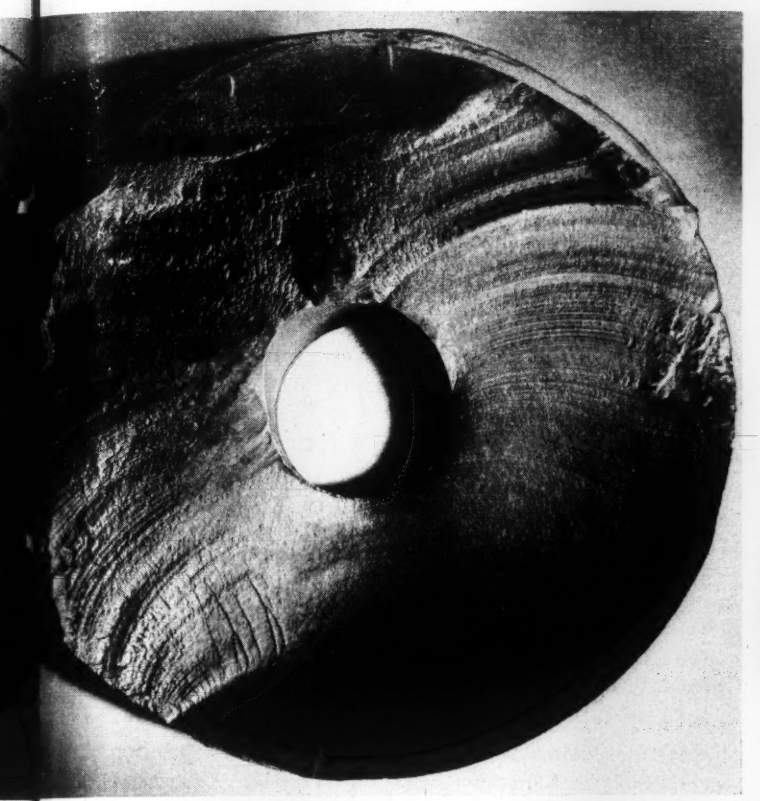
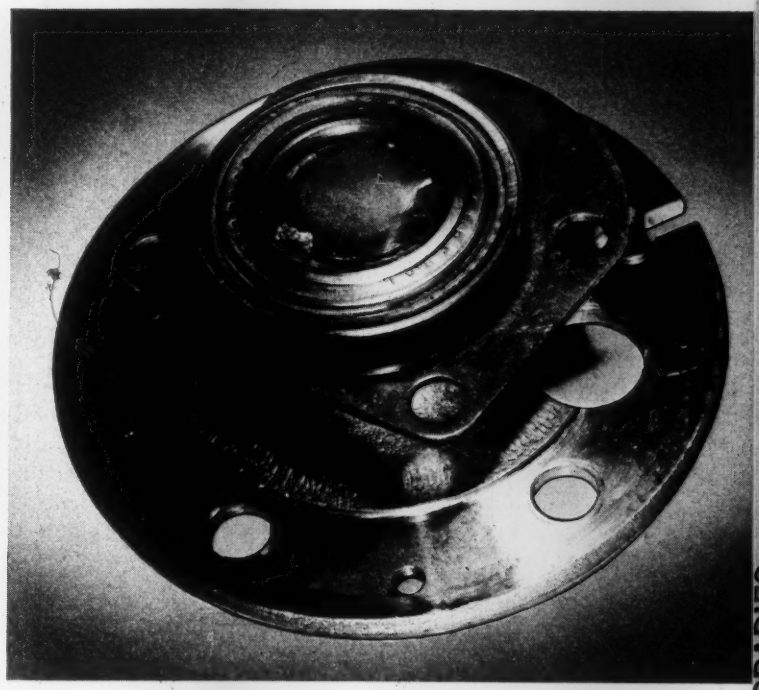


Fig. 39—Stop marks on this fatigue fracture of a locomotive driving axle show where each overload occurred and the course taken by the fracture



Photo, courtesy Harry Ferguson Inc.

Fig. 40—Above—Failure of this rear axle shaft occurred in a laboratory test after 1,439,000 cycles at a load four times normal. No stop marks appear on the surface of the fracture because of cyclically uniform loading

Photo, courtesy Republic Steel Corp.

Fig. 41—Fatigue failure of this crankshaft produced a fracture with beach marks, the radial lines at top of fracture. They mark the junction of separate cracks originating from different points on the periphery

until so much of the total cross section has been cracked through that the remaining solid metal can no longer sustain even a single load application, and so ruptures suddenly.

The part illustrated in *Fig.* 36 failed in fatigue after approximately one year of service. As the photograph indicates, the fatigue crack originated at one corner of the tang and spread in a generally peripheral direction until the area of the remaining sound metal in the opposite corner was so small that the tang broke off completely at the final load application. This condition could be alleviated by a more generous fillet at the base of the tang.

First phase of this fatigue mechanism takes many cycles and often a long period of time. As a result, the progressive portion of the fatigue fracture is smoothed by the continual rubbing that takes place between the two faces of the crack. That portion of the metal which failed instantaneously, however, has a rough, grainy surface.

STOP MARKS: The smooth area of the fractured surface is further identified in practically all service failures by what are known as fatigue *stop marks.* Unless the stop marks have been obliterated during the final phase of the fracture, they will make quite evident the rate of progress of the fatigue crack.

Finding practically no indication of stop marks in

fatigue fractures obtained in the laboratory, Bacon[1] attempted to synthesize them. He first tried sudden load reduction in the middle of the fatigue test but found no marks to indicate the interruption. Next he tried the same procedure but applied an increased load for a short duration in the middle of the test. By this latter means he was able to produce the marks synthetically. It would therefore appear that stop marks are caused by sudden increases in the applied load. *Fig.* 39, a service fatigue failure of a locomotive main axle, shows stop marks indicative of sudden overloads.

In *Fig.* 40 is shown a laboratory fracture of a rear axle shaft subjected for 1,439,000 cycles to a load equivalent to four times normal. It is significant that this fracture indicates no stop marks; the cyclical load application was uniform during the entire test. This appearance is typical of fractures obtained in laboratory fatigue tests.

Speed of Crack Propagation

Study of the shape of border lines between zones and of the location and size of the instantaneous crack allows conclusions to be drawn on the progress of failure. It appears to be true that the speed of crack penetration is proportional to the square of the depth of the crack. The absolute rate of advance, however, depends on the intensity of stress.

CRACK ORIGIN: Since a fatigue failure is by definition the result of the gradual progression of a crack, there arise the questions: Why and where does the crack originate? In general, most fatigue cracks start at or near an outside surface, inasmuch as stresses are apt to be greatest at the surface. This is particularly true for torsion and bending. The particular surface point at which the crack starts will depend upon the geometry of the machine member and the surface conditions.

Any notch or other stress raiser is an excellent candidate for a fatigue crack nucleus. Countless examples of notch-initiated fatigue failures can be shown to illustrate this statement. Fillets that are too small, threads, oil holes, keyways, and similar surface irregularities are all potential nuclei of fatigue failures. Typical examples of fatigue failures originating from such surface irregularities are shown in *Figs.* 36, 38 and 41.

It should be noted that a machine member with no geometric irregularities can still be subject to fatigue failure. This circumstance can be explained by a review of the origin of fatigue cracks. Although a machine member may be free of geometric irregularities on a gross scale—having no shoulders, threads or sudden changes in cross section—it may still contain a great number of stress raisers on a smaller scale. Such stress raisers are tool marks, scratches, identification stamp marks, or any of various inherent discontinuities in the material itself, such as inclusions or quenching cracks. The smoother the surface and the more uniform the material, the greater the fatigue strength because of fewer susceptible fatigue crack nuclei. Ground and polished pieces, for instance, demonstrate higher fatigue strength than rough turned pieces, which in turn are superior to as-forged pieces.

BEACH MARKS: Magnification of stress in the region of a stress raiser, such as a small fillet or a peripheral groove, causes the fracture to be initiated at several points almost simultaneously. As these individual cracks propagate, they join to form the final fracture. Shown in *Fig.* 41, the lines of juncture of these cracks are known as *beach marks.*

STRESS SHADOW: An interesting phenomenon in the study of fractures is illustrated in *Fig.* 42. The stress concentration resulting from the fillet is somewhat mitigated by the keyway in the heavy section. The zone of reduced stress is known as a *stress shadow.* The fatigue crack will start at the side opposite the shadow and progress toward it. Usually the shadow zone will be intersected by the plane of the final rupture.

For further study, additional photographs of fractured parts and discussions of pertinent factors will be found in the sources listed in References 2 to 6.

The next article of this series will show how to recognize failures resulting from bending and axial loads.

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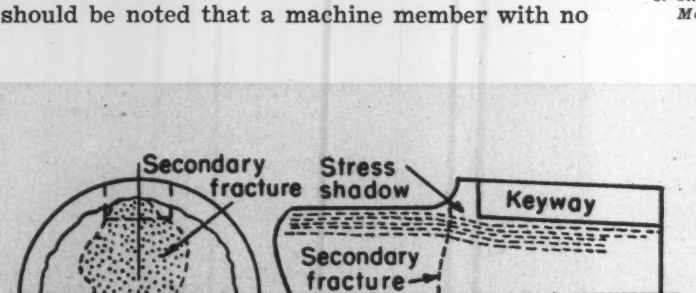

Fig. 42—This diagram by Meyercordt illustrates the stress-shadow phenomenon. In the zone near the keyway the stress concentration due to the fillet is relieved. The fatigue fracture avoids this area although the secondary fracture occurring at the final load application generally passes through the stress shadow

MEN OF MACHINES

Reinout P. Kroon

Emil P. Knapp

John F. Miller

Reinout P. Kroon, engineering manager of the aviation gas turbine division, Westinghouse Electric Corp., Philadelphia, Pa., has been awarded the Spirit of St. Louis Medal of the ASME for leadership in the development of the first American design of a turbojet power plant for aviation service, making possible increased plane speeds, higher altitude limits and more efficient airplane design; and for his technical leadership in many fields which have contributed to the advance of American aviation. Mr. Kroon is a graduate of the Zurich Graduate Technische Hochochule in Switzerland. He joined the Westinghouse Corp. in 1931 and in 1945 was appointed to his present position.

♦

Square D Company announces the appointment of **Emil P. Knapp** as chief product engineer. Mr. Knapp has been with the Kollsman Instrument Division of the Square D Co. since 1934, first as project engineer and for the past five years as assistant chief engineer. He received his engineering training in Europe and has the degree of mechanical engineer from the State Engineering College at Esslingen, Germany. The functions of the Kollsman product engineering department have been expanded considerably due to the steadily increasing number of new product designs.

♦

John F. Miller has been elected vice president of Illinois Tool Works, Chicago, Ill. Mr. Miller joined Illinois Tool Works last September as manager of the company's tool division. In his new position, Mr. Miller will continue to manage the engineering of the standard and special metal cutting tools produced by Illinois Tool Works. He will also direct the development and marketing of the company's new line of graphic inspection equipment. This equipment was designed to record precision inspection results in easily interpreted graph form.

♦

Aluminum Co. of America announces the appointment of **Dr. Kent Van Horn** as associate director of research. He will be transferred to the central headquarters of Alcoa's research organization at New Kensington, Pa. Dr. Van Horn is a graduate of Case Institute of Technology and one of the nation's outstanding metallurgists. In 1944, he was elected pres-

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ident of the ASM. Dr. Van Horn is also a member of the AIME and ASTE as well as the author of numerous technical articles and textbooks.

♦

Dr. John Ray Dunning has been appointed dean of the School of Engineering at Columbia. Beginning in 1932 he pioneered in some of the first neutron experiments conducted in the United States, and directed the building of the first Columbia University cyclotron in 1936. Dr. Dunning has contributed in important degree to other engineering developments in atomic work.

♦

R. L. Peaslee has joined the Wall Colmonoy Corp., Detroit, as development engineer. Mr. Peaslee will be in charge of the development of Colmonoy brazing alloy as a stainless and alloy steel brazing material.

♦

J. Edward C. Anderson has joined the Pesco Products Division of Borg-Warner Corp., Bedford, Ohio as a project engineer.

♦

H. H. Morgan, vice president of the Robert W. Hunt Co., has been awarded a certificate by the American Standards Association for his outstanding service in standardization. The award recognizes his service in policy-making in support of the standards movement as a means of advancing the national economy.

♦

Hewitt A. Gehres, executive vice president and director of engineering of the Cooper-Bessemer Corp., received the honorary degree of Doctor of Science from the Grove City College, Grove City, Pa.

♦

C. E. Davies, secretary of the ASME, has received the honorary degree of Doctor of Engineering from the Drexel Institute of Technology.

♦

Paul H. Davey Jr., has been elected vice president in charge of production of the Davey Compressor Co. A graduate of Yale College of Engineering, Mr. Davey has been production manager for the company for the past two years.

♦

Professor Harry J. Loberg has been appointed director of Sibley School of Mechanical Engineering at Cornell University.

♦

Westinghouse Electric Corp. has awarded the company's highest honor—the Westinghouse Order of Merit to **Dr. Arthur M. Wahl,** engineer at the Westinghouse Research Laboratories, Pittsburgh, Pa. Dr. Wahl was honored for his exceptional skill in applying the principles and results of stress analysis to the design problems of the company and for his ability in developing solutions to complex technical problems.

♦

C. S. Davis Jr., has been promoted to the presidency of the Borg-Warner's Norge Heat Division in Detroit. Mr. Davis was formerly vice president and general manager of this division.

♦

Allegheny Ludlum Steel Corp. announces the appointment of **George I. Bottcher** as chief engineer. Mr. Bottcher joined the engineering department of Allegheny in 1941 and was appointed assistant chief engineer in 1947. **J. F. Chapman** and **R. E. Smith** were appointed as assistants to the chief engineer.

♦

DeVilbiss Company of Toledo, Ohio announces the appointment of **Norman F. Smith** as plant superintendent. For the past three years, Mr. Smith has been chief process engineer with the Pyle National Co.

♦

Louis E. Theisen was appointed vice president in charge of engineering of the Metal Products Corp.

♦

Midvale Co. announces the appointment of **M. Worth Acker** as chief engineer. Mr. Acker is now in charge of all engineering for the company.

♦

Hyatt Bearings Division of General Motors Corp., announces the appointment of **H. R. Gibbons** as chief engineer in charge of product design, research, application and service engineering. Mr. Gibbons is a graduate of Stevens Institute of Technology.

♦

Selden T. Williams, vice president of Scovill Manufacturing Co. Inc., was elected president of A. Schrader's Son Inc.

♦

Harold W. Zipp has been appointed executive engineer, directing all engineering administration and staff activities for the Wichita Division, Boeing Airplane Co.

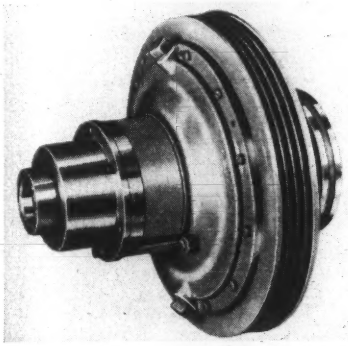
♦

Bendix Aviation Corp., announces the appointment of **Capt. Howard Thomas Orville** as director of engineering for the Friez Instrument Division. Captain Orville is a pioneer in the development of equipment and instruments for obtaining upper-atmosphere data. He graduated from the U. S. Naval Academy and received his master of science degree from MIT.

NEW PARTS AND MATERIALS

For additional information on these new developments see Page 179

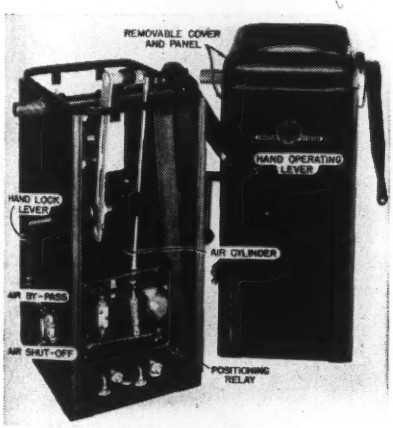
Magnetic-Particle Clutch



Linking action of dry magnetic mixture in magnetic field between driving and driven parts transmits torque in Magneclutch controllable coupling. Magnetic mixture is composed of iron particles and flake graphite. Magnetic field is established by current flowing through coil. By varying current, degree of clutching can be controlled. Design and operating features include extremely fast response, minimum wear on torque transmitting surfaces, torque at zero slip, large maximum to minimum torque ratio, small control power requirements and easy adaptation to remote control. Manufacturer: Vickers Inc., Vickers Electric Div., 1815 Locust St., St. Louis, Mo.

For additional information circle MD 1 on Page 179

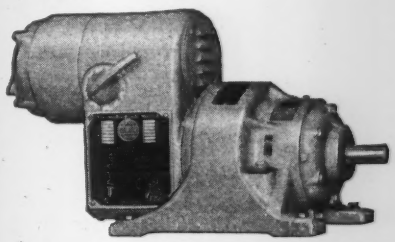
Air-Operated Control Drive



This completely enclosed air-operated drive makes it possible to parallel characteristics of a driven device with those of other equipment to secure a unified system. Drive consists of double-acting air cylinder, positioning relay and manual operating wheel or lever. Positioning relay incorporates cam that determines rotation of control drive position to control loading pressure. Three cam shapes are included with each drive and can be used interchangeably to obtain positions which are proportional to first power, square or square root of control loading pressure. Torque requirements of 280-400 ft-lb are accommodated by unit with 6-in. bore and 8-in. stroke, while size with 8-in. bore and 16-in. stroke is usable for torque up to 1500 ft-lb. Manual operation is simplified by air-supply shut-off valve, bypass valve, hand lever or wheel and clamping device. Manufacturer: Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.

For additional information circle MD 2 on Page 179

Variable-Speed Drive



Available in ¼, ⅓, ½ and ¾-hp sizes, type VA-GD variable-speed motor unit combines Varidrive which permits instant change of speed in ratio of 1:10 and heavy duty Syncrogear for increasing torque. Speed ranges are from 10 to 10,000 rpm. Frame of Varidrive is cast aluminum. Dial control permits instant speed change by turning handle less than one revolution to obtain any speed change. Slippage is prevented by incorporation of Autotaut belt tensioner which automatically compensates for various loads. Gears of Syncrogear unit have hardness of 45—50 Rockwell C. Quick inspection of interior is afforded by cover plate. Manufacturer: U. S. Electrical Motors Inc., 200 E. Slauson Ave., Los Angeles 54, Calif.

For additional information circle MD 3 on Page 179

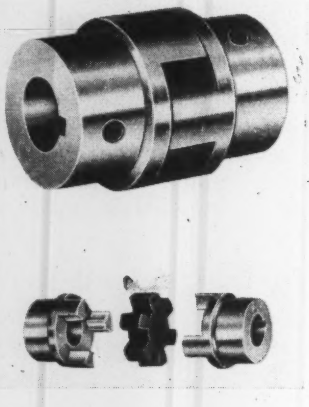
Hermetically Sealed Relays

Line of hermetically sealed relays has application where humidity or dust conditions present hazard to continuous operation. Wide variety of contact combinations and alternating and direct-current operating voltage ranges are available. Supplied in four standard mounting arrangements: lug header type, Army-Navy connector type, octal-plug type and

screw terminal type. Illustration shows series 335 direct-current hermetically sealed model. Manufacturer: Guardian Electric Mfg. Co., 1621 W. Walnut St., Chicago 12, Ill.

For additional information circle MD 4 on Page 179

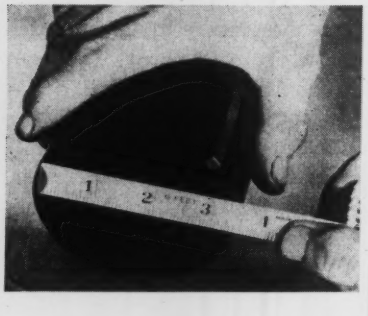
Flexible Coupling



Resilient synthetic rubber insert or spider of this flexible coupling is held in position by concave jaw surface. Tendency of rubber to extrude or break is eliminated. Coupling is vibration free in operation and self-adjusting for misalignment. It is available in sizes for shafts from ½ to 1⅝ in. and in ratings from fractional to 25 hp. Standard keyways are provided. Manufacturer: Gerbing Mfg. Co., Roto Cone Variable Speed Pulley Div., Northbrook, Ill.

For additional information circle MD 5 on Page 179

Heavy-Wall Plastic Pipe



Flexibility, light weight and resistance to rust and electrolytic corrosion are features of Carlon EX heavy-wall plastic pipe. Usable at temperatures from −70 to 140 F, pipe has high impact strength, burst pressures up to 260 psi and resistance to suction collapse. Dielectric strength is 500 v per mil. Furnished in 2, 2½, 3, 4 and 6-in. standard diameters, the pipe can be coupled quickly and easily by means of molded plastic fittings which permit plastic-to-plastic or plastic-to-metal connection. Internal and external sleeve-type fittings can be utilized in many installations. Red stripe insures positive identification. Manufacturer: Carter Products Corp., 10303 Meech Ave., Cleveland 5, O.

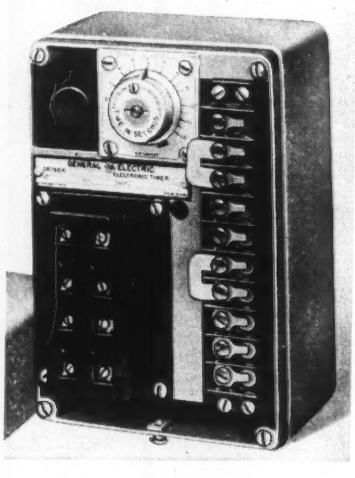
For additional information circle MD 6 on Page 179

Vinyl Insulating Compound

Underwriters' Laboratories approved for use as electrical tubing at 105 C and for use as appliance wiring insulation at 90 and 105 C, Geon plastic 8630 can be supplied in colors for easy circuit identification. Features include resistance to high temperature aging, good abrasion and deformation resistance, flexibility at high and low temperatures and smooth glossy surface. It withstands oils and chemicals, is noncorrosive to copper and has high dielectric strength which permits thin insulation walls. Manufacturer: B. F. Goodrich Chemical Co., 324 Rose Bldg., Cleveland 15, O.

For additional information circle MD 7 on Page 179

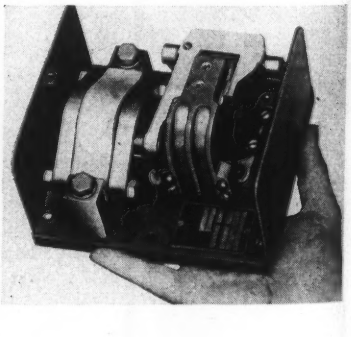
Industrial Electronic Timer



Automatic operation, limit and sequence timing are possible with this electronic timer. Timing ranges of 0.06 to 1.2-seconds, 0.6 to 12-seconds and 6 to 120 seconds can be obtained. Relay armature is only moving part and standard electronic tube is employed. Load requirements are 15-amp inrush, 10-amp carry and 5-amp break. Detachable back plate mounts directly on rigid conduit or any flat surface. Space between terminal board and cover provides adequate room for front-connected wiring. Dual contacts permit control of two operations on same time schedule. Manufacturer: General Electric Co., Control Div., Schenectady 5, N. Y.

For additional information circle MD 8 on Page 179

Magnetic Air Valve



Rotary design of this magnet-operated air valve permits high capacity and fast operation in an extremely small-size control unit. Direct-acting magnet and low inertia of rotary valve motion provide continuous operating speeds of 300 cycles per minute and up to 600 cycles per minute on intermittent duty. Designed as a four-way, four-port ⅜-in. valve for control of a double-acting air cylinder, the unit can be converted to three or two-way operation by plugging proper ports. Pressures up to 100-psi can be handled. Valve body and rotor are Manganese bronze; all magnet parts are hardened for durability. Rotor is pressure-balanced to minimize bearing wear

and has spring-loaded vanes to insure positive seal. Valve can be mounted in any position. Manufacturer: Square D. Co., 4041 N. Richards St., Milwaukee 12, Wis.

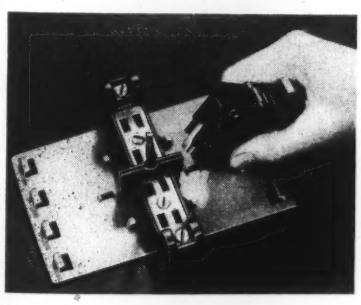
For additional information circle MD 9 on Page 179

Potentiometer Controller

Temperature, speed, flow, pressure, static strain loads, voltage, etc., of any industrial equipment can be controlled with extreme accuracy and speed by Potentiotrol potentiometer controller. Measurements can be obtained with any primary element producing electric impulse. Device is essentially a combination of an electronic control circuit with balanced or potentiometric type measuring system. Instrument is recommended for installations where use of "suppressed ranges" and null type measuring system are desired. Selected span is spread over large open scale and 9½-in. slide wire for readability from distant points as well as for close control. Manufacturer: Wheelco Instruments Co., Chicago 7, Ill.

For additional information circle MD 10 on Page 179

Air Circuit Breaker

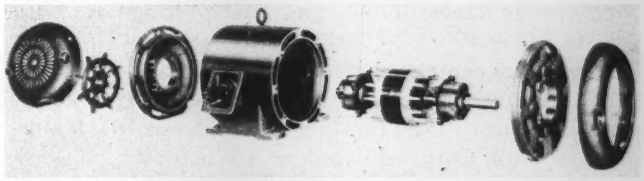


Consisting of two springs and handle assembled in case, Stab-Lok air circuit breaker is of unit-pole construction and has thermal-magnetic overload. It can be inserted in enclosing case in same manner as fuse. Unit is approved by Underwriters' Laboratories. Manufacturer: Federal Electric Products Co., 50 Paris St., Newark, N. J.

For additional information circle MD 11 on Page 179

Totally Enclosed Motors

Supplied in ratings from 5 to 250 hp in both standard type EP and explosionproof type JP designs, these cast iron frame motors are especially suitable for outdoor installations or where corrosion is a problem. Frame, endplates, blower shield and conduit box are cast iron. Blowers used on type EP motors are malleable iron and those employed by type JP motors are nonsparking bronze. On all except largest ratings,



blower design permits operation of motors in either direction of rotation. Rotor squirrel-cage is cast aluminum and is dynamically balanced. Other design features include sealed leads, cartridge type ball bearings, protected stator laminations and varnish-treated windings. Manufacturer: Wagner Electric Corp., 6400 Plymouth Ave., St. Louis 14, Mo.

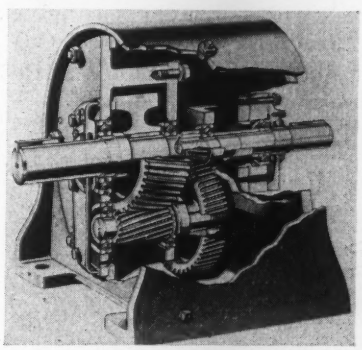
For additional information circle MD 12 on Page 179

Rotary Snap Switch

Compact and totally enclosed, No. 2800 rotary snap switch is offered in a wide variety of circuit sequences including double and single pole series-parallel, and on and off circuits. It is rated at 15 amp 125 volts ac and can be supplied with four positions or limited position. Switch body is 1¾ in. high, 1⅜ in. wide, and ¾-in. deep. It is usually furnished with type S shaft, but special shafts can be supplied. The switch is adapted to such applications as panel boards, instruments, computing mechanisms, air-conditioning equipment, etc. Black, white or colored plastic indicating rotary switch handles can be furnished. Manufacturer: Ark-Les Switch Corp., 51 Water St., Watertown 72, Mass.

For additional information circle MD 13 on Page 179

Speed Reducers



Designed for a wide variety of power transmission requirements, these concentric and right-angle type units are suitable for use as speed reducers or as speed increasers. Both units are enclosed in all-steel housings and are supplied in an approximate range of 1 to 50 hp. Efficiency is 96 to 98½ per cent at full load.

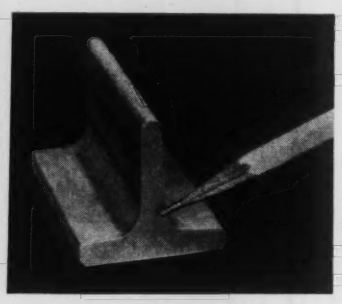
Type CB right-angle reducer is available with output shaft in either horizontal or vertical position. Output speeds from 1 to 350 rpm are possible with

input speeds up to 1750 rpm. Ratio range is from 5.7:1 to 515:1.

Type C concentric shaft unit has a ratio range of 1.5:1 to 970:1. It delivers from 1 to 1170 rpm at input speeds up to 1750 and higher if necessary. Manufacturer: Falk Corp., Milwaukee 8, Wis.

For additional information circle MD 14 on Page 179

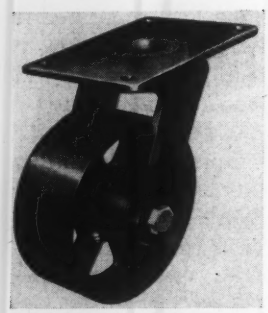
Nickel Alloy T-Sections



Having application as structural parts for furnaces and other high temperature equipment, hot-rolled Inconel is supplied in equal T-sections measuring 1½ by 1½ by ¼-in. in lengths to 15 ft. Weight is approximately 2¾ lb per ft. Section can be easily welded, riveted or otherwise joined. Structurals of same size can also be supplied in nickel and Monel where tonnage demands are sufficient. Manufacturer: International Nickel Co., Huntington, W. Va.

For additional information circle MD 15 on Page 179

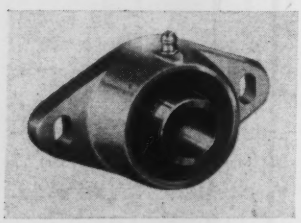
Forged-Steel Casters



Designed for heavy-duty service on racks, dollies and portable equipment, 500 series steel-forged casters are flame-hardened to produce hard wearing surface in ball raceways for fast swiveling action. Casters are equipped with 2½-in. wide 4, 6, 8 or 10-in. diameter wheels in either swivel or rigid types. Models are available with nicro-steel, Durastan plastic, molded-on rubber, or V-trac wheels. Top plate is 5¼ by 7¼ in., with bolt hole spacing of 3⅝ by 5¼ in. Manufacturer: Rapids-Standard Co., Dept. C-500, 342 Rapistan Bldg., Grand Rapids 2, Mich.

For additional information circle MD 16 on Page 179

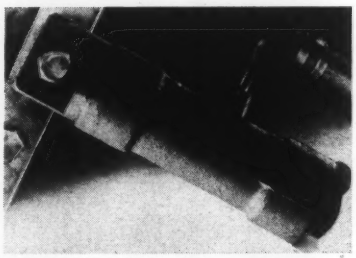
Ball Bearing Flange Unit



Featuring metal two-bolt style housing, FED series ball bearing flange unit for use on small shaft equipment handles light loads at normal speeds. Precision internal self-aligning ball bearings compensate for shaft deflections and mounting inaccuracies. Labyrinth-type seal minimizes drag and wear. Unit is locked to shaft by headless set screw and standard grease fitting is provided for supplementary lubrication. Manufacturer: Ahlberg Bearing Co., 3025 W. 47th St., Chicago 32, Ill.

For additional information circle MD 17 on Page 179

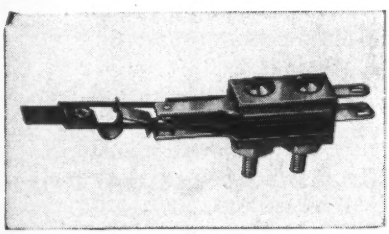
Pneumatic Vibrators



Featuring full-power start and minimum air consumption, series 76 pneumatically operated vibrators deliver 2200 to 6000 impacts per minute, depending upon piston diameter and line pressure. Air consumption ranges from 4½ to 8 cfm at 80-psi line pressure. Piston diameters vary from ½ to 2½ in. Bronze alloy cylinder liner insures permanent piston lubrication. Manufacturer: SPO Inc., 6534 Grand Division Ave., Cleveland 5, O.

For additional information circle MD 18 on Page 179

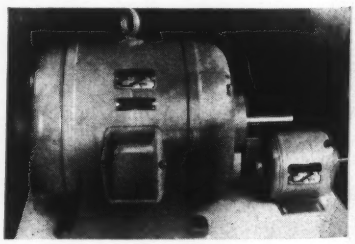
Snap-Action Switch



Designed for use in conjunction with relays and limit switches, this snap-action switch will accommodate high inductive loads with minimum arcing. Main operating blade is tempered spring steel and can be furnished to operate at various pressures. Conventional switch can be used for single-pole single-throw or single-pole double-throw requirements. Special types can be furnished on specification. Manufacturer: Comar Electric Co., 3148 N. Washtenaw Ave., Chicago 18, Ill.

For additional information circle MD 19 on Page 179

Quick-Stopping Motor

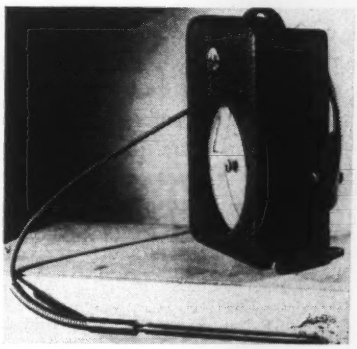


Fundamentally an induction motor with standard squirrel-cage rotor, this unit is designed for applications requiring quick stopping. Stator employs, in addition to its running winding, a single-phase alternating current winding for electrodynamic braking. When motor stopping mechanism is actuated, circuit to running winding is disconnected and circuit completed between power

source and braking winding. Under influence of braking winding, retarding force several times that of full load torque is applied to rotor. As motor approaches complete stop, braking force becomes negligible. Motor is obtainable in sizes from 1/20 to 10-hp NEMA size 286 in single and polyphase types. Manufacturer: Standard Dayton Corp., P. O. Box 1001 Dayton 1, O.

For additional information circle MD 20 on Page 179

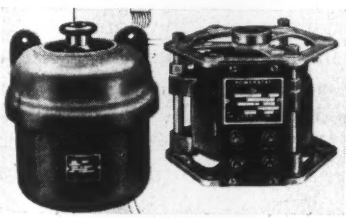
Temperature-Pressure Recorder



Temperature and pressure can be recorded under variety of changing conditions by this 6-in. chart recorder. Instrument can be converted from temperature to pressure recorder and changed in recording range. Pressure system ranges cover from 0 to 5000 psi and temperature system ranges cover from —40 to 1000 F. Bulb of instrument is supplied with mercury, gas or vapor filling according to use. Recorder can be flush or wall mounted; legs and handle are available to make device portable. Manufacturer: Gotham Instruments Div., American Machine & Metals Inc., 149 Wooster St., New York 12, N. Y.

For additional information circle MD 21 on Page 179

Variable Transformer



Continuously - adjustable source of alternating - current voltage for hazardous areas is supplied by this exposionproof Powerstat variable transformer. The totally enclosed unit will withstand internal gas or vapor explosions and will prevent ignition by internal spark or flash of vapor or gas surrounding enclosure. Underwriters' Laboratories approved for Class 1, Group D service, the transformer is supplied in 115 and 230-v ratings for single-phase operation. Manufacturer: Superior Electric Co., Hannon Ave., Bristol, Conn.

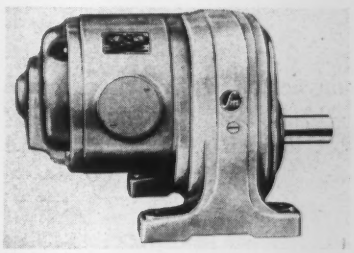
For additional information circle MD 22 on Page 179

Industrial Hydraulic Fluid

Excellent lubricating qualities, nontoxicity and flame resistance are features of OS-16 industrial hydraulic fluid. Ester-base compound will not corrode bearings or other metal parts and is nonconductor of electricity. Field tests in hydraulically operated mining equipment at 1500 psi have indicated its value as power transmission medium. Manufacturer: Monsanto Chemical Co., St. Louis 4, Mo.

For additional information circle MD 23 on Page 179

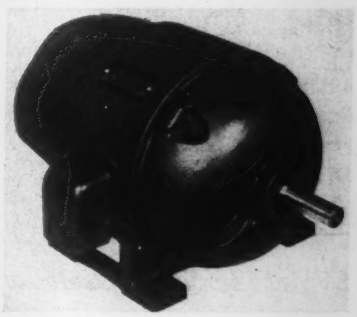
Motorized Speed Reducers



Now available in splashproof design for horizontal floor mounting, Slo-Speed geared electric power drives are suited for locations which are subject to direct streams of water or other liquid. Built with single or double reduction gears, motors provide speeds from 780 down to 20 rpm in ratings of ½ to 15-hp. One-piece cast iron stator frame and end bells with integrally cast baffles are construction features. End bell design permits free passage of ventilating air through motor while preventing splashing liquids from reaching interior. Seals on shaft ends insure positive protection of bearings. Manufacturer: Sterling Electric Motors Inc., Los Angeles, Calif.

For additional information circle MD 24 on Page 179

Repulsion-Induction Motor

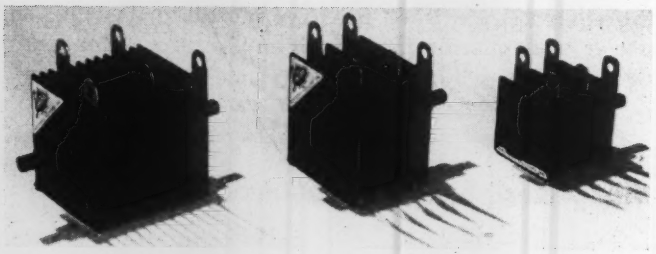


Combining high starting torque of repulsion type motor with constant-speed characteristics of induction motor, type SCR single-phase motor is available in 5, 7½ and 10-hp 1800-rpm ratings. Latter two models use 230 v while 5-hp type operates on 115/230 v. All have cast-iron frame and are of open, dripproof type. To insure good alignment between stator and rotor, ball bearings are mounted directly in end shields. Cooling is provided by single large-diameter cast aluminum fan. Manufacturer: General Electric Co., Small & Medium Motor Divs., Schenectady 5, N. Y.

For additional information circle MD 25 on Page 179

Dry Disk Rectifiers

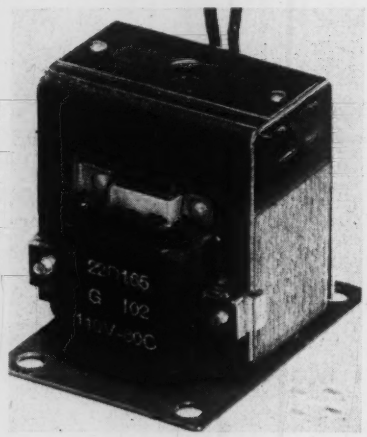
Designed for use in small battery and trickle chargers as well as in other low-voltage direct-current supply applications, Lektron magnesium-copper sulphide dry disk rectifiers are offered in three sizes. Capacities are 2-4, 4-6 and 6-10 amp. Unaffected by atmospheric conditions, rectifiers can be cooled by



convection. Instant operation at temperatures from −40 to 284 F is possible. Manufacturer: Electronic Rectifiers Inc., 2104 Spann Ave., Indianapolis, Ind.

For additional information circle MD 26 on Page 179

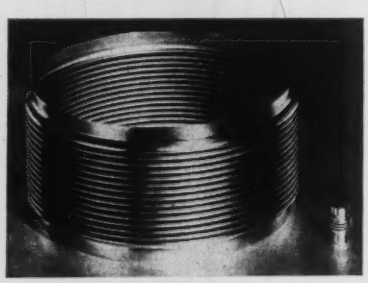
Short-Stroke Solenoid



Featuring a cast Permafil coil assembly and one-foot Flamenol leads, this push type solenoid is available in four models with strokes of ⅛ and 5/16-in. and two stacking thicknesses. The solid block around coil is resistant to moisture, oil, shock and is vibrationproof. The push mechanism consists of a stainless steel push pin brazed to the plunger. A Neoprene pad fastened to the backstop absorbs shock of return stroke. Manufacturer: General Electric Co., Control Divs., Schenectady 5, N. Y.

For additional information circle MD 27 on Page 179

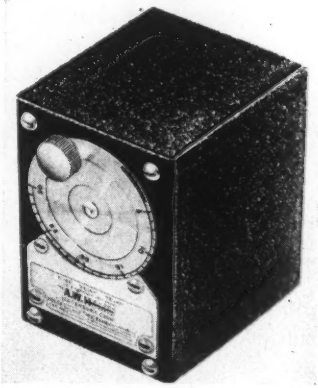
Stainless Steel Bellows



Especially suited for applications in control and instrumentation, these stainless steel bellows can be used where problems of pressure, high and low temperatures and corrosion are factors. They are manufactured in single and multiple-ply construction in various lengths, dependent upon requirements. Bellows or bellows assemblies with standard fittings or fittings to customer's specifications can be supplied. Fittings can be attached by electrical circular seam welding to insure leakproof assemblies. Manufacturer: Chicago Metal Hose Corp., 1308 S. Third Ave., Maywood, Ill.

For additional information circle MD 28 on Page 179

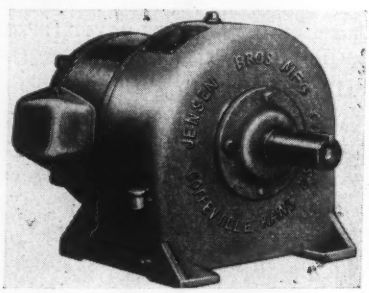
Time Adjustment Dial



Usable with line of time delay relays, these adjustable dials can be surface or flush mounted. Time can be changed easily, and setting locks positively. Flexible design of relay allows for addition of radio-interference filter and a governed motor on direct current models. Clutch-holding contacts for push button start can be supplied on either alternating or direct current models. Manufacturer: A. W. Haydon Co., Waterbury, Conn.

For additional information circle MD 29 on Page 179

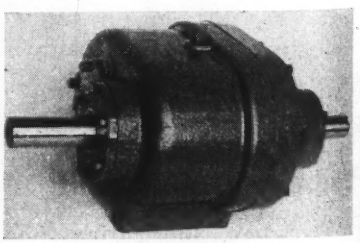
Speed Reducer



Available in a variety of sizes from ½ to 10 hp, GeartorK speed reducer consists of a system of planetary gears mounted in a cast semisteel frame. The above sizes are also supplied with double-reduction gear trains. The electric motor is bolted directly to the frame; speed reducer and motor form a compact, easily mounted unit requiring only slightly more space than the motor itself for installation. Flange-mounted models can be obtained for special applications. Construction features include oil-bath lubrication, machined and shaved gears and main shaft mounted on antifriction bearings. Manufacturer: Jensen Bros. Mfg. Co., Coffeyville, Kan.

For additional information circle MD 30 on Page 179

Two-Speed Transmission



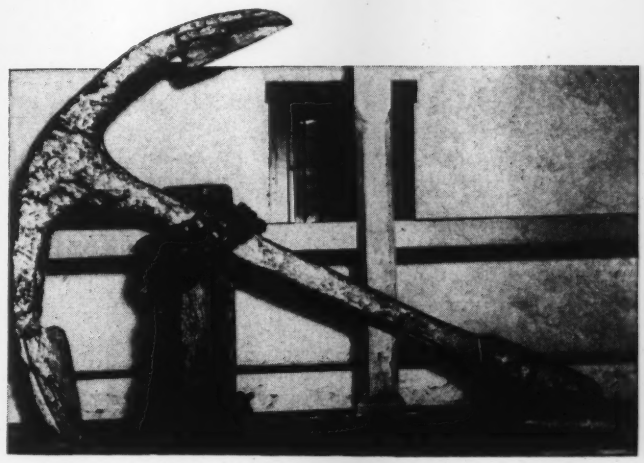
Any preselected ratio from direct motor speed to 6:1 reduction can be obtained with this two-speed automatic transmission. It has a 20-hp capacity with a minimum input speed of 900 rpm. The mechanism consists of two over-running clutches so arranged that by reversing the motor, two speeds are provided without changing direc-

Corrosion of Bearings-1

ALL bearing materials should be fabricated to meet requirements of corrosion resistance. These requirements are of two distinct types. Corrosion resistance prior to installation and corrosion resistance after installation.

Prior to installation, bearings may corrode during shipment or in storage. The corroding medium usually is water or water vapor due to occasional exposure to high humidity or actual condensation of water where cold metal is brought into a warm room under such conditions that the temperature of the metal is below the dew point. In some rare cases corrosion may be caused by certain gases present in the atmosphere. These instances should be given individual study in order to arrive at the proper preventive. The resistance of the bearing metals to corrosion due to water or prior to installation is of a different order than corrosion resistance under operating conditions. In this first treatise we will limit the discussion to the former.

The approximate tendency to corrode in the presence of water is shown in figure No. 1, under room temperature conditions. Steel is the most sensitive of all the materials used and precautions have to be taken to protect the steel against rusting. The amount of protection required depends on the expected time of exposure to atmospheric conditions. If steel-backed bearings are to be placed in service shortly after manufacture, then a light film of oil will suffice. For periods of protection up to six months an electro-deposit of tin or a heavier oil coating is used. For periods of protection up to two years an electro-deposit of tin, plus a light grease is used. For longer periods or for overseas shipment a heavy grease is often used.



An excellent example of resistance to corrosion is this old bronze anchor recently dredged up from Monterey Bay, California, where it had lain in *salt water* for over 300 years

A protective oil is required for lead-base babbitt for storage periods longer than six months.

Both steel and lead-base babbitt are characterized by pitting when corrosion due to water vapor takes place. A few pits will not harm either of these materials but extensive corrosion may result in undesirable dimensional changes.

Copper-lead will corrode because the lead phase is exposed but the lead is isolated in small areas and not subject to the more extensive attack of lead-base babbitt.

Tin base babbitt, bronze and aluminum may stain on exposure to the atmosphere but damage due to corrosion under these conditions is practically unknown.

The corrosion of bearings under service conditions is due to entirely different factors than under storage conditions.

Service conditions are discussed in data sheet . . . Corrosion-2.

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

●

This bearing sheet data is but one of a series. You can get the complete set by writing to—

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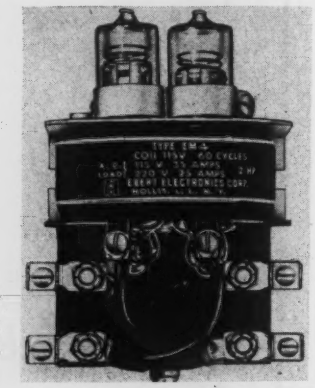
Relative Comparison of Corrosion

Fig. 1 Tendency of bearing material to corrode in the presence of water vapor at room temperature.

tion of rotation of output shaft. Speed control is provided by means of a push button, limit or timing switch or other electrical means. Manufacturer: Western Mfg. Co., 3400 Scotten Ave., Detroit 10, Mich.

For additional information circle MD 31 on Page 179

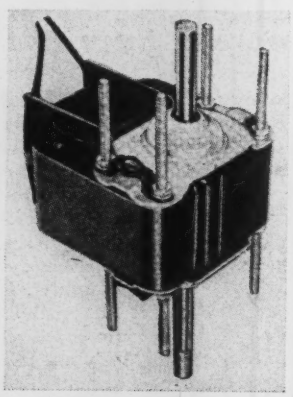
Mercury Plunger Relays



Incorporating two standard EMT tubes in one coil structure, type EM-4 two-pole mercury-plunger relay handles loads up to 35 amp at 115-v ac. Measuring 4⅜ in. high, 3⅜ in. wide and 2¼ in. deep the unit is available with contacts normally open, normally closed or with one contact normally open and the other normally closed. Latter arrangement permits instantaneous switch from one circuit to another. Features are hermetic dustproof construction with heavy tungsten contacts in hydrogen-filled glass tube. Contact is made from mercury to mercury when internal magnetic plunger is energized. Manufacturer: Ebert Electronics Corp., 185-09 Jamaica Ave., Hollis 7, N. Y.

For additional information circle MD 32 on Page 179

Shaded-Pole Motor

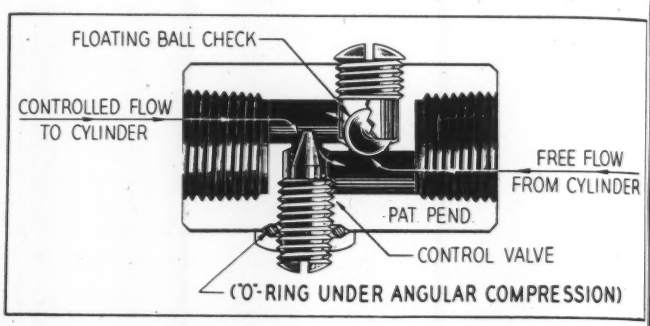


Means for accurate positioning of gears with motor shaft is construction feature of redesigned Raytheon 350 shaded-pole, 1/40 to 1/7-hp, 3000-rpm motors. Assembly utilizes two centerless-ground tubular-steel dowels which can be extended beyond bearing brackets of motor to furnish indexing position in relation to motor shaft. Other improvements include addition of stamped aluminum bracket, larger bearing and remodeled bearing spring. Latter permits more end thrust as bearing rides against bracket instead of spring. Manufacturer: Russell Electric Co., Chicago, Ill.

For additional information circle MD 33 on Page 179

Flow Control Valve

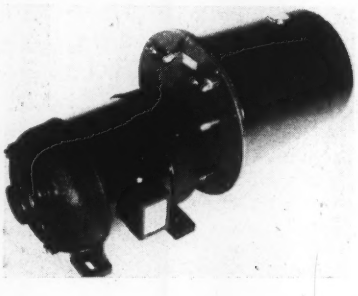
Improved Pneu-Trol flow control valve for air and hydraulic cylinder speed control employs Dryseal pipe threads to insure tight joints and to eliminate air or fluid leakage at pipe-valve connection. Floating ball check provides effective and sensitive control.



Valve is available in ⅛, ¼, ⅜, ½ and ¾-in. sizes. Compact body design permits consolidated piping arrangements. Manufacturer: Pneu-Trol Devices Inc., 3122 N. California Ave., Chicago 18, Ill.

For additional information circle MD 34 on Page 179

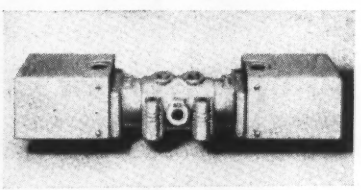
Hydraulic Pumping Unit



Available with either 5-pint or 3-gallon reservoir, improved Midget hydraulic pumping units have outputs of 0.1 to 2.8 gpm. Type with 6½-pint reservoir is available on special order. Pump pressures are preset according to specifications. Reservoir is mounted on open, totally enclosed or explosionproof motor; pump, check valve and relief valve are enclosed. Both reservoir and bulkhead are aluminum, and units with open motors have aluminum motor frames. Pressure and return line connections are for ⅜-in. flared tubing, but ¼-in. pipe thread or hose coupling connections can be supplied. Pumping units are adaptable for portable applications. Manufacturer: John Dusenberry Co., 271 Grove Ave., Verona, N. J.

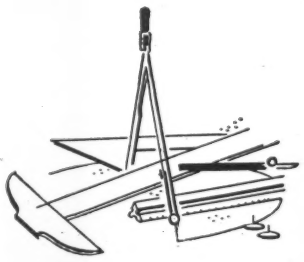
For additional information circle MD 35 on Page 179

Four-Way Hydraulic Valve



Employing a hydraulically balanced piston which has integral spool and stem, this series 5500 double-solenoid four-way valve is designed for use as a semiautomatic electric control of a hydraulic circuit. Two solenoids, continuous-duty push type, are located at each end of the valve body. Entire piston surface is hardened, ground and lapped with special grooving to minimize possibility of binding under high pressure conditions.

Valve is offered in double solenoid operated type with either standard action or spring centered action; single solenoid operated with spring return, and



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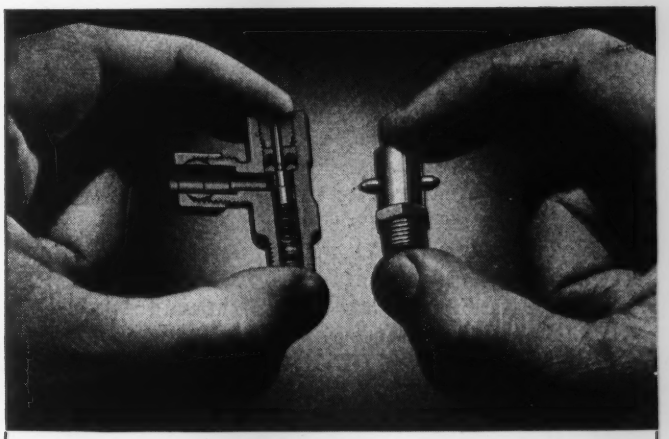
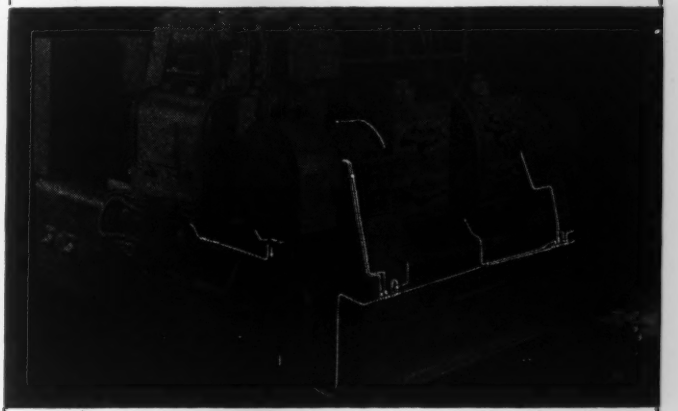


1 Alemite "Midget" System–slightly larger than a pin-type fitting, this Alemite "Midget" measuring valve is readily applicable to all types of heavy, light and precision machinery. Its simple, compact design makes it especially suitable wherever space limitations are encountered. Mounted on or near the bearings, the Alemite "Midget" delivers a measured amount of oil or grease from one central point. Can be installed for either manual or automatic operation.

2 Alemite Single Line Terminating System...applicable to heavy or light machines, precision machines, outdoor installations. It is a single lubricant line, hydraulically operated system for oil or grease. Valves can be installed on or near each bearing. A measured amount of lubricant, large or small, is conveyed to each bearing from one central point while machine operates. An indicator signals when job is done. Choice of manual or automatic operation.



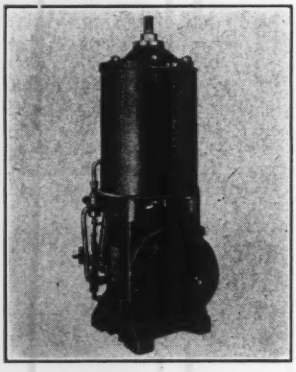
3 Alemite Single Line, Reversing System is ideal where excessive dirt and moisture is a problem. The system is completely enclosed, fully hydraulic. Each valve delivers a metered amount of oil or grease to two bearings. The lubricant flows *progressively* to each bearing while the machine is producing . . . an indicator at the point of introduction signals when lubrication is completed. Designed for manual or automatic operation.

solenoid pilot operated.

Designed for pressures up to 1500 psi, it is available in ¼, ⅜, ½, ¾ and 1-in. sizes for 110, 220, and 440-v 50/60 cycle operation. Other current specifications can be furnished. Manufacturer: Rivett Lathe & Grinder Inc., Brighton 35, Boston, Mass.

For additional information circle MD 36 on Page 179

Hydraulic Actuator

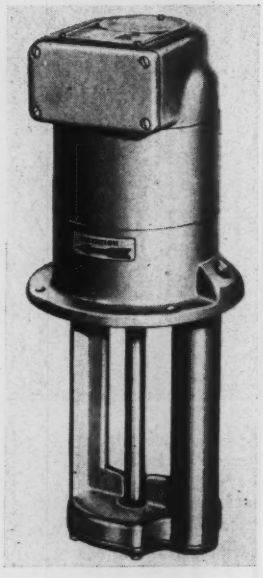


Hydractor, a compact, self-contained, double acting hydraulic actuator, delivers positive straight line motion with cushioning action at both ends of stroke. Assembly consists of motor, pump, hydraulic cylinder, sealed-in charge of synthetic hydraulic fluid, and solenoid operated control valve. Base, foot, or clevis mounted, the Hydractor provides hydraulically powered linear motion in a packaged unit just as a motor provides rotary motion.

Available in three basic force capacities and fifteen models, the 1000 pound and 2000 pound models have linear speeds ranging from 3 ips to 12.1 ips, and the 4000 pound units operate at 2.2 ips to 6.7 ips. Standard stroke is 12 inches. Longer or shorter strokes, adjustable speed controls, load sustainer and other modifications are available to make the Hydractor a versatile machine. Manufacturer: Benjamin Lassman & Son, Oliver Building, Pittsburgh 22, Pa.

For additional information circle MD 37 on Page 179

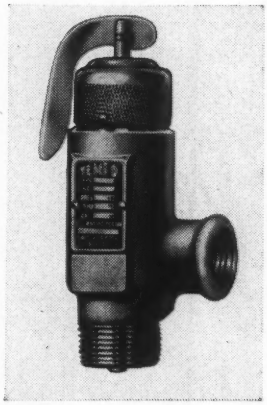
Coolant Pump



Impeller housing, spacer supports, mounting flange and lower motor end bell of Gusher model 6 P3 coolant pump consist of one casting. One spacer support is cored to provide discharge passage from pump through plate flange and to simplify mounting in tank or machine bed plate. Equipped with permanently lubricated precision ball bearings, 1/10-hp 3450-rpm driving motor has precision ground shaft extending through entire length of unit. Rotating assembly is dynamically balanced by electronic process. Manufacturer: Ruthman Machinery Co., Cincinnati 2, O.

For additional information circle MD 38 on Page 179

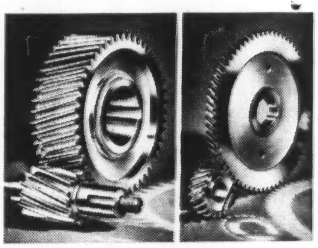
Relief Valve



Available with either short or long stem and with or without try levers, type TD Tempo automatic reseating relief valve has standard relief pressure setting of 125 psi and temperature setting of 210 F. Other settings can be specified when ordered. All parts are accessible from top, and valve can be cleaned without being taken off line. Features include nonsubmerged bellows assembly that forestalls liming and snap-action design that insures self-cleaning of working parts each time valve operates. Manufacturer: A. W. Cash Valve Mfg. Corp., Decatur 80, Ill.

For additional information circle MD 39 on Page 179

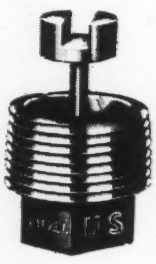
Standardized Gear Sets



Pre-engineered for known life expectancy and horsepower capacity, Duti-Rated gears can be selected by procedure comparable to selecting antifriction bearings. By combining file-hard tooth surfaces with ductile cores, these gear sets have load carrying capacities and wear life comparable to those of commercial gearing but feature considerable reduction in size and weight. Gears can be obtained in 200 combinations with ratios from 1.5 to 9.3 and center distances from 2.7 to 15.5 in. Ratings range from fractional up to several hundred horsepower, and gearing is available in four basic pinion constructions and two gear types. Manufacturer: Foote Bros. Gear & Machine Corp., Chicago 9, Ill.

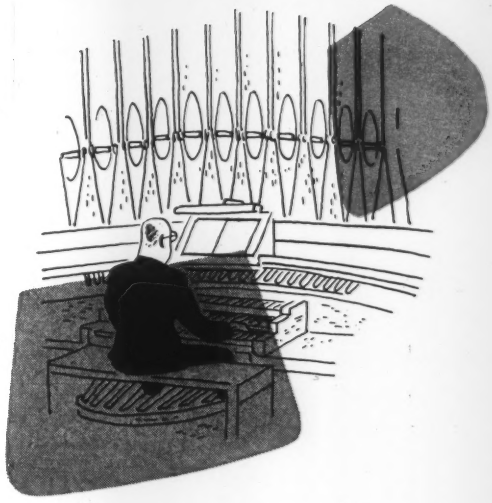
For additional information circle MD 40 on Page 179

Magnetic Plugs



Made of sintered Alnico, this magnetic plug will remove ferrous metal particles from moving lubricant. By varying length of nonferrous supporting stem, magnet can be placed anywhere in lubricant stream. Cleaning is accomplished by air blasting. Having tractive force of 11 oz, plug employs Dryseal thread which requires no pipe compound for installation. Plug is available in ¼ to 1½-in. sizes with standard square, countersunk or special design head. Manufacturer: U. S. Plug & Fitting Co., 501 High Ave., Cleveland, O.

For additional information circle MD 41 on Page 179



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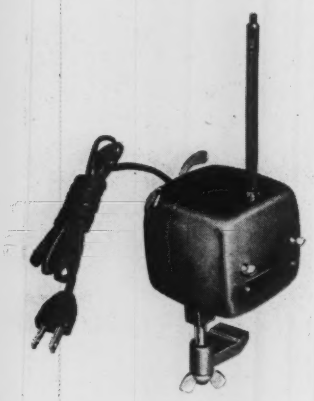
EQUIPMENT

For additional information on this new equipment see Page 179

Drafting Lead Sharpener

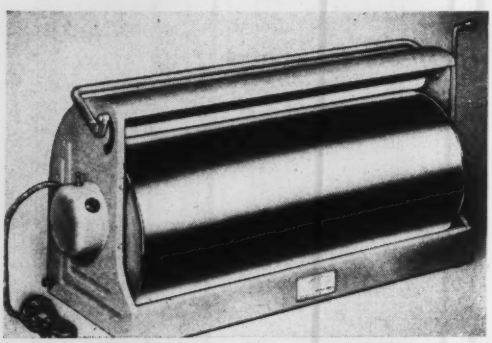


For drafting or photo retouching work, this precision sharpener contour grinds uniform round and chisel points on standard leads in mechanical holders from 1/32-in. below chuck to tip. Operating on 115-v 60-cycle ac, sharpener has lever for adjustment to desired point size. Manufacturer: Rajean Mfg. Co., 906 Kingston Ave., Racine, Wis.

For additional information circle MD 42 on Page 179

Portable Print Dryer

Low-cost Rotomaster portable print dryer can be used for photographs, blueprints, Vandykes and photocopies. Dryer employs 23 by 28-in. mirror-finish stainless steel drum and built-in squeegee to simplify application of prints to drying surface. Two-heat element

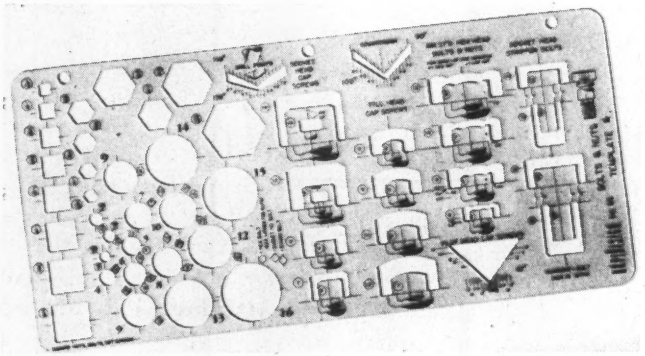


maintains constant temperature according to setting and will not scorch prints. Unit dries full load in from 4 to 8 minutes and requires 300 w at 115-v ac. Manufacturer: Lott Mfg. Co., 51 Parker Ave., Jamestown, N. Y.

For additional information circle MD 43 on Page 179

Bolt and Nut Template

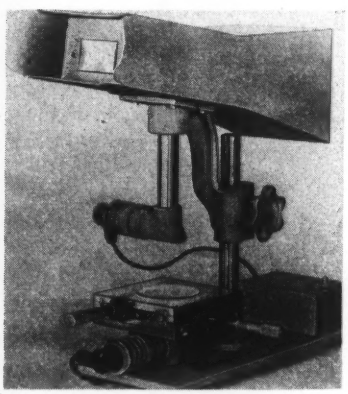
This No. 85 drafting template contains cutouts for socket head cap screws; American Standard hex-head



bolts and nuts; square bolts, nuts and setscrews; socket head stripper bolts and fillister cap screws; flat head cap screws; and drill point and countersink angles. It is made of 0.030-in. double-cured mathematical quality plastic with printing on matte surface to minimize wear. The template measures 5 by 10-7/16 in. Manufacturer: Rapidesign Inc., Box 592, Glendale, Calif.

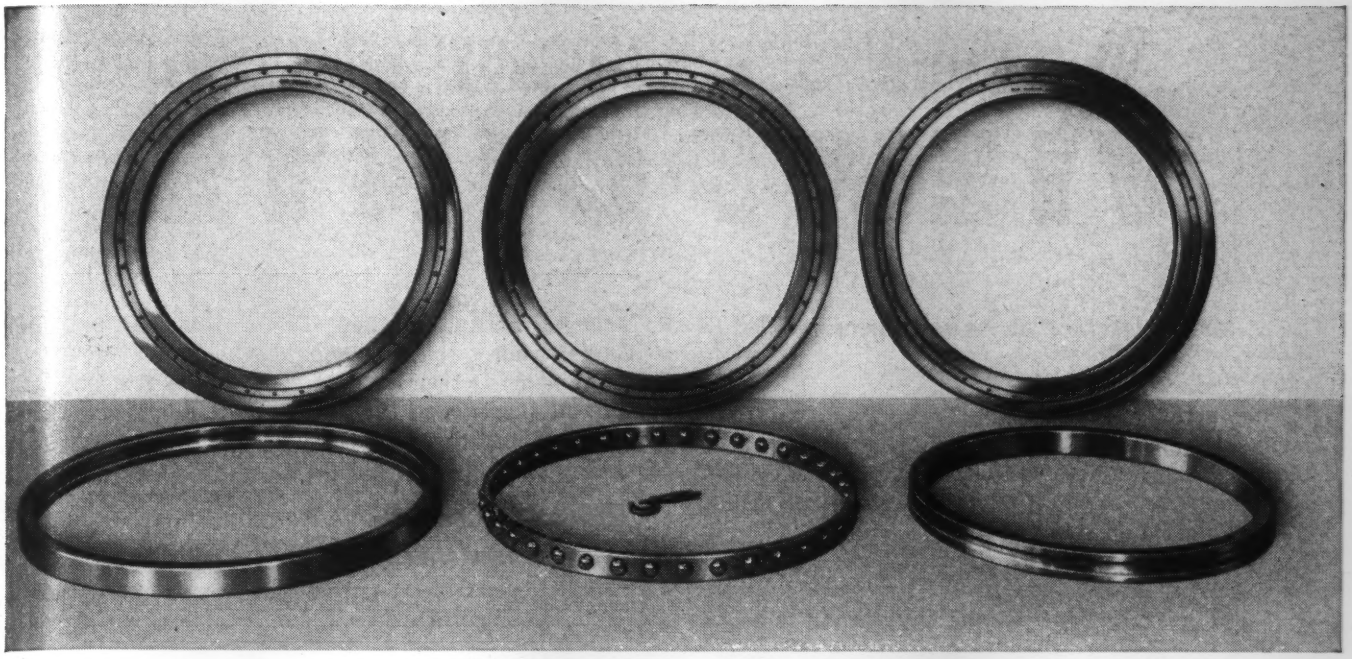
For additional information circle MD 44 on Page 179

Optical Projector

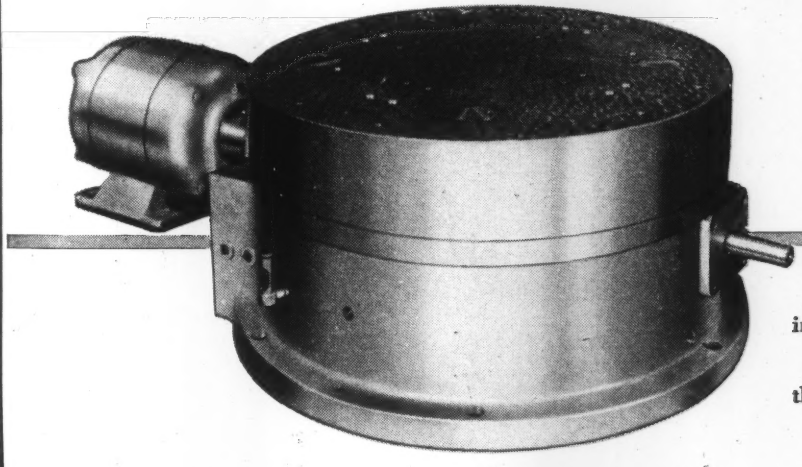


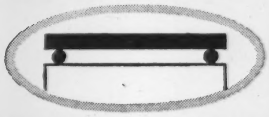
In addition to serving as a measuring instrument with readings in increments of 0.0001-in. in two planes at 90 degrees, this universal projector provides both contour and front-surface projection of subject being measured. It is available in standard magnifications of 10, 20, 30, 45 and 60 power as well as any special magnification up to 120 power. Screen is 6 in. square and is supplied with single vertical and horizontal cross lines. Additional lines or layouts can be added quickly by means of scribe and vernier height gage. Measuring stage has 1-in. travel in two planes at 90 degrees and is 5 in. square. Since projection hood can be raised or lowered, parts up to 5 in. high can be measured. Manufacturer: Stocker & Yale, Association Bldg., Marblehead, Mass.

For additional information circle MD 45 on Page 179



BIG KAYDON PRECISION BEARINGS PROVIDE PRECISION SUPPORTS FOR NEW "RIM-BALL" INDEX TABLES





A special KAYDON 4-point Ball Bearing, pre-loaded in assembly of the new line of Rehnberg-Jacobson "Rim-Ball" Index Tables, provides support underneath and just inside the outer edge all the way around. Designed to take heavy thrust and radial loads in any direction, this bearing completely prevents any tendency of the table to tip or distort, even slightly.

All six table sizes, ranging from 16″ to 42″, in the new line of R-J "RIM-BALL" Index Tables, are equipped with the type of KAYDON Special Ball Bearings shown above . . . a 35″ outside diameter bearing, for instance, is pre-loaded in assembly on the largest table.

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PROFESSIONAL VIEWPOINTS

". . . . further information on bevel gear article"

To The Editor:

In the article "Bevel Gear Trains" by Mr. Paul Grodzinski appearing in the March 1950 issue of the magazine, I have found some items that I cannot follow through by computation, by assumptions made, or by statements in the article:

1. The left-hand side of Equation 5 does not follow from Equation 4.
2. The algebraic sign of the right-hand side of Equation 13 does not follow from Equation 12.
3. The right side of Equation 15 should read $b + a \cos \gamma$.
4. The incorrect sign in Equation 12 leads to discrepancies in Equations 19, 20 and 24.

—RAYMOND S. KUKFA
Troy, N. Y.

" . . . errors in the formulas"

To The Editors:

Unfortunately, there are a few errors in the formulas of my article. For Equation 5 I have to accept the responsibility, but Equations 12 and 15 seem to have become incorrect through typographical errors. After correcting Equations 5 and 12, Mr. Kukfa's objections resolve themselves without comment.

However, I have found a further misprint in Equation 8, and would therefore be glad if you could bring the following corrections to the attention of your readers. Equation 5 should read

$$2CP = N_1/\sin \alpha_1 = N_2/\sin \alpha_2 = N_3/\sin \alpha_3$$

Equation 8 should read

$$\tan \alpha_1 = \sin \gamma_1/[(N_1/N_2) + \cos \gamma_1]$$

Equation 12 should read

$$b \sin \gamma_1 - a \sin \gamma_2 = \cos \gamma_1 \sin \gamma_2 - \sin \gamma_1 \cos \gamma_2 = \sin (\gamma_2 - \gamma_1)$$

and Equation 15 should read

$$r \cos \varepsilon = b + a \cos \gamma.$$

—PAUL GRODZINSKI
London, England

". . . . cut-and-try bevel gear solutions"

To The Editor:

Having just had occasion to refer to the article on "Bevel Gear Trains" by Paul Grodzinski, that appeared in your March 1950 issue, I submit herewith a cut-and-try solution of a problem of the kind Mr. Grodzinski is trying to simplify. Symbols used conform to ASA B6.5-1949, Letter Symbols for Gear Engineering.

Problem: Determine pitch cone angles for the gears and the position of the intermediate shaft for a bevel gear train of three gears, given

$$\Sigma = 75°$$

$$M_G = 4.70 \text{ to } 4.75$$

$$N_P(Min) = 20$$

The number of teeth on the idler is to be approximately the mean of other two, and is to be prime to both:

$$N_G = (4.70 \text{ to } 4.75) \times 20$$

$$= 94 \text{ or } 95. \text{ Use } 95$$

$$N_I = (20 \times 95)^{0.5}$$

$$= 43.5. \text{ Use } 43$$

The following trials and final solution are on the basis of 1 diametral pitch, the first three being by slide rule, while the last three were done with a calculating machine.

Trial No.	1	2	3
A_o	80	79	79.8
γ_P = arc sin $(20/2A_o)$	7.2°	7.28°	7.21°
γ_I = arc sin $(43/2A_o)$	15.6°	15.8°	15.67°
γ_G = arc sin $(95/2A_o)$	36.5°	37.0°	36.6°
Σ $(\gamma_P + 2\ \gamma_I + \gamma_G)$	74.9°	75.88°	75.15°

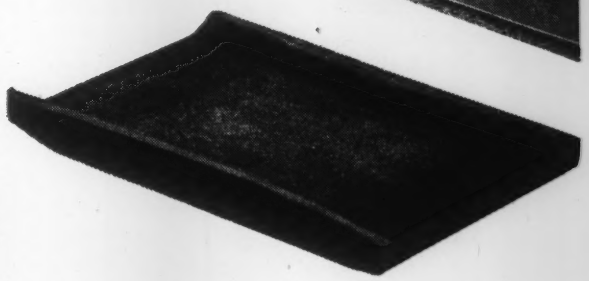
Trial No.	4	5	6
A_o	79.900	79.800	79.789
sin γ_P	.125156	.125313	.125330
sin γ_I	.268086	.269423	.269460
sin γ_G	.594493	.595238	.595320
γ_P	7°11′23″	7°11′56″	7°11′59″
γ_I	15°36′36″	15°37′48″	15°37′56″
γ_G	36°28′35″	36°31′46″	36°32′ 8″
Σ_1 $(\gamma_P + \gamma_I)$	22°47′59″	22°49′44″	22°49′55″
Σ_2 $(\gamma_I + \gamma_G)$	52° 5′11″	52° 9′34″	52°10′ 4″
Σ $(\Sigma_1 + \Sigma_2)$	74°53′10″	74°59′18″	74°59′59″

The first trial value (80) of the cone distrance A_0 was chosen after assuming that the pitch cone angle of the gear would be about one-half of the 75-degree shaft angle. The value of A_0 used in trial No. 6 was determined by extrapolation from the values used in trials 4 and 5, assuming that within the limited range covered by these three trials Σ varies inversely as A_0.

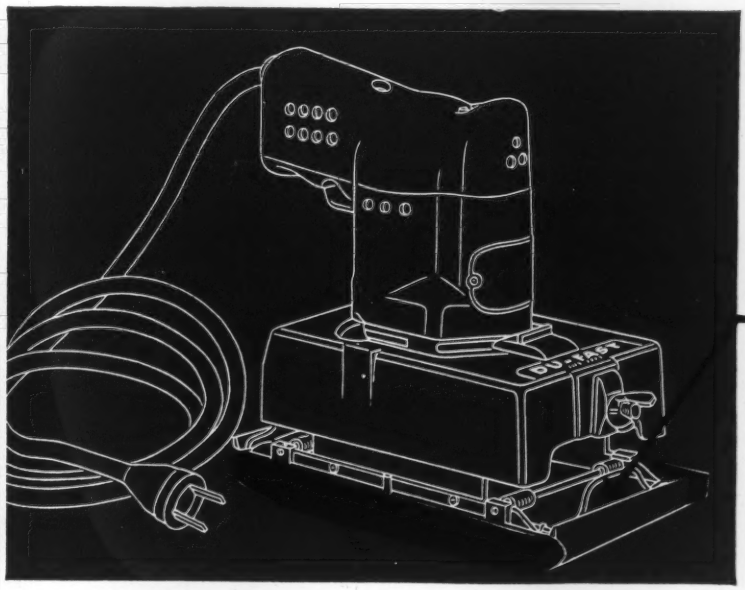
In this solution there has been no need to construct the auxiliary chart, *Fig.* 3 of the article referred to. Neither has there been any occasion to resort to the use of an infinite series. The difference between the sum of pitch cone angles and shaft angle is one second against the author's 20, and the 180-degree shaft angle, instead of being a special case, is simply another of the same kind. Extending this method to a train involving an internal bevel would present no

to an imaginative product designer . . .

Spongex® resilience



yielded this idea



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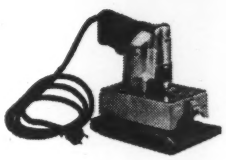

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great difficulty. I think that many will agree that the method outlined here is as simple as or simpler than the method given in Mr. Grodzinski's article.

—HENRY C. GRAY
The Falk Corp.
Milwaukee, Wis.

". . . springs with minimum solid height"

To the Editor:

In the article on "Springs" in the March issue of MACHINE DESIGN the author proposes a method based on calculus to find a spring with minimum solid height for given conditions. He applies this method to an example in which he finds a "minimum" solid height of 3.2141 inches. It is easy to show that springs with less than 3 inches solid height can be designed which meet all the conditions of the problem; data for two such springs are given below. This indicates that there must be some error in the author's method.

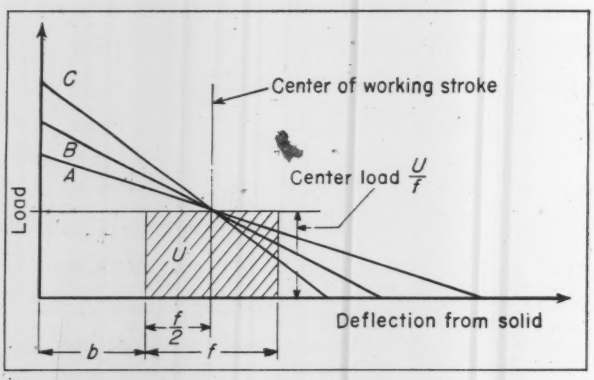
Three of the many springs which meet the conditions of the problem are as follows:

Spring Number	Wire Dia. d (in.)	Outside Dia. $D+d$ (in.)	Active Coils N	Rate k (lb/in.)	Solid Load F_s (lb)	Solid Stress S (psi)	Solid Height H (in.)
1	0.1875	1.4381	15.142	60	225	133098	3.2141
2	0.207	1.44	11.75	120	300	133000	2.85
3	0.1875	1.44	12.5	73	241	141000	2.72

Spring Number 1 is the author's solution. The values given to five or six significant figures are his. The rounded values for rate and solid load are calculated from his data.

Number 2 is a spring of commercial wire size which absorbs the same energy in the working stroke, has the same total clearance at the end of the working stroke (more between adjacent coils) same solid stress, same outside diameter, but shorter.

Number 3 is made of the same wire and coil diameter as the author's spring and meets all other conditions, but the stress is arbitrarily raised another 8000 psi, just as the author raised the stress at the end of his calculations from the assumed 125,000 to 133,000. The result is of course a much shorter spring. This spring is shown to bring out the importance of stress in problems of this type.



Load-deflection curves of three springs which absorb the same energy U in the working stroke f. Spring B with solid load twice the fixed center load, absorbs least total energy from free to solid. Spring C is stiffer than B, absorbs more total energy, but may be shorter. Spring A is more flexible than B, absorbs more total energy and must be longer

The mathematical errors which result in a spring longer than necessary are hidden in Equation 12 of the article. The variable which should have been differentiated is Nd, not N, and this variable should have been differentiated with respect to the independent (unknown) variable d, not with respect to r which is fixed for all practical purposes by the permissible spring diameter.

Going through the differentiation and the necessary calculations after eliminating these errors, we have arrived at mathematically correct results; these results are not shown because the fine edge of calculus is here used to cut a knot which can be unraveled easily with ordinary arithmetic. In problems of this type physical understanding is much more essential than mathematical complications.

Proposed Method of Attack

First decide on the highest permissible stress; it is most important. Comparing Springs 1 and 3, a 6 per cent increase in stress gave more than 15 per cent decrease of solid height. If the load is static, the permissible stress can be raised by using preset springs. If the load is frequently repeated, shotpeening will permit higher stresses and smaller springs.

Then choose the "configuration". A nest of two springs will decrease the needed space. Special flattened spring wire will also decrease the space requirements. We assume that a single spring of round wire is desired.

Next make trial calculations with a few wire diameters, d. The chosen stress and the given hole diameter determine the maximum permissible load F_s for each wire (author's Equation 1). The maximum load and the center load, F, which is the same for any spring which meets the conditions of the problem, together with the fixed distance from center of working stroke to solid, determine the necessary rate.

The center load, F, is the load at the center of the work stroke. Any spring which absorbs an energy, U, in a working stroke, f, will have the same center load, $F=U/f$, at the center of the working stroke provided it has a constant rate, k, which is assumed in the article. This is shown in the accompanying load-deflection curves.

In the author's example $U=300$ in.-lb, $f=2$ in., $F=150$ lb, deflection from end of working stroke to solid $=b=0.25$ in. and therefore the rate must be made $k=(F_s-150)/1.25$ lb/in.

If this rate turns out to be higher than $2U/f^2$, 150 lb/in. in the example, the energy U could be absorbed in a stroke smaller than f or the full strength of the assumed wire will not be used. This will happen if the wire diameter, d, is assumed too big for the conditions of the problem. Rate k, wire diameter d and hole diameter determine the necessary number of active coils (author's Equation 3) and the solid height is thus determined.

A number of possible springs can be calculated by these elementary methods and the shortest spring chosen in little time. It should be kept in mind that even with the aid of higher mathematics a couple of check calculations of this type will be necessary be-

(*Concluded on Page* 188)

How to cut costs when drafting revisions are necessary

* **A case history based on the experience of the Virginia Department of Highways**

1

Long-lasting intermediates are assured. In a "permanence test" made by the Virginia Dept. of Highways, an Autopositive print was left on a roof top for 36 days. During this time this photographic intermediate was exposed to 200 hours of sunlight . . . 6.88 inches of rain. Despite all of this abuse it was declared "good as new."

Proof, indeed, that "Autopositives" will stand up under less trying, normal conditions . . . will remain intact in the files year after year . . . ready to produce sharp, clean blueprints whenever needed.

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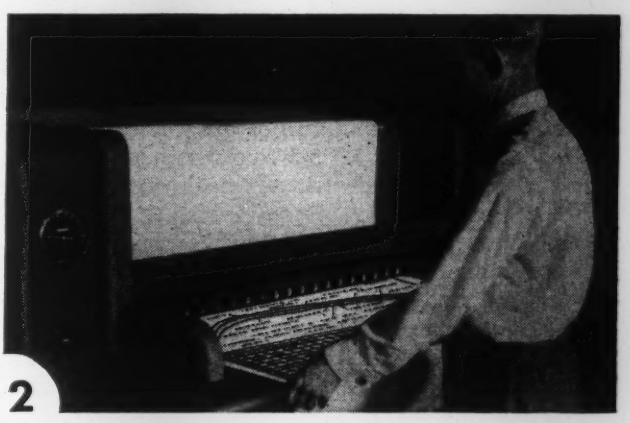

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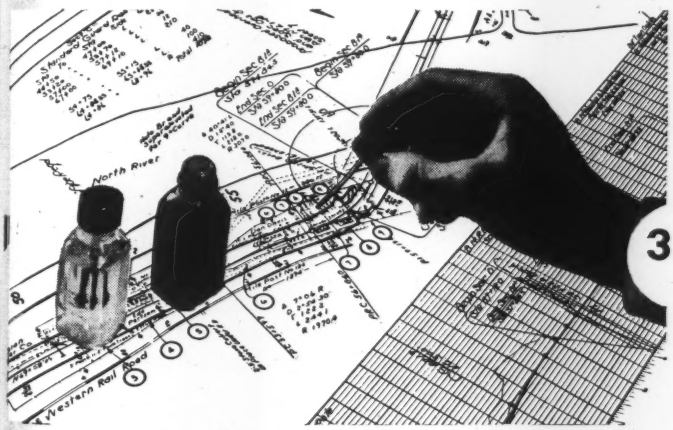
Photographic intermediates are produced at a new low cost. When "Autopositive" is used, positive photographic intermediates are produced *directly* without a negative step, without darkroom handling. Maximum efficiency is realized by the Virginia Dept. of Highways because its "Autopositives" are turned out automatically . . . in a continuous blueprint machine, which can be converted readily for Autopositive production.



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Drafting revisions are easily made. Unwanted details—such as existing right of ways—are removed quickly from "Autopositives" with corrector fluid. Then the proposed right of ways are drawn in with pencil or ink. Thus, new masters—prepared without costly redrafting—are ready to turn out the blueprints needed for county supervisors and resident engineers.

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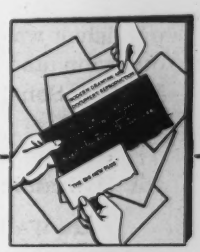

19

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Resistance Welding
Designing, Tooling and Applications

By Wallace A. Stanley, consultant, Progressive Welder Co.; published by the McGraw-Hill Book Co. Inc., New York; 329 pages, 8½ by 11 inches, clothbound; available through MACHINE DESIGN, *$7.50 postpaid.*

Based on material secured directly or indirectly from the American Welding Society, various manufacturers of resistance welding equipment, and professional associations and journals, this book discusses spot welding, projection welding, seam welding, and flash-butt welding. Spotlighting modern methods and developments, the material is presented in three successive stages: Basic information on resistance welding, the application of basic process details in designing welded parts, and the use of this background and design data in tooling and production.

Comprehensive and well-illustrated, the book should prove a ready reference for the engineer in the design of welded machine parts for economical and efficient production.

Kent's Mechanical Engineers' Handbook

Twelfth edition, in two volumes: **Design and Production** *volume, 1611 pages, edited by Colin Carmichael, editor of* MACHINE DESIGN; **Power** *volume, 1409 pages, edited by J. Kenneth Salisbury, general engineering and consulting laboratory, General Electric Co.; published by John Wiley & Sons Inc., New York; clothbound, 5⅝ by 8⅝ inches; available through* MACHINE DESIGN, *$8.50 per volume, postpaid.*

In the review of this handbook, which appeared in the July issue of MACHINE DESIGN, the name of the publisher was inadvertently omitted. We regret the omission and apologize to our readers and to John Wiley & Sons.

Oil Hydraulic Power and Its Industrial Applications

By Walter Ernst, vice-president and director of engineering, The Commonwealth Engineering Co. of Ohio; published by the McGraw-Hill Book Co. Inc., New York; 366 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$6.00 postpaid.*

This book should prove of value to design engineers considering the application of oil hydraulic power to drive machine components. The author's presentation is divided into sections on generation, utilization, transmission, control, and application. Each of these divisions includes detailed discussions of the components involved and their integration into a system.

The pages devoted to the generation of power, for example, deal with rotary pumps—the type used almost exclusively in present design. Principal representatives are gear pumps, vane pumps, and radial and axial plunger pumps. Models of these types made by various manufacturers are covered thoroughly—detailed design features, performance characteristics, limitations, advantages, and disadvantages. Design studies involving displacement, power, capacity, piston diameters, length of stroke, and other factors are presented. Sufficient theory for basic reference appears in the opening chapters of the book.

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Fine-Pitch Straight Bevel Gears ASA B6.8—1950: This new standard was developed to cover generated straight bevel gears, of 20-diametral pitch and finer, for all shaft angles and with the numbers of teeth equal to or greater than 16/16, 15/17, 14/20, or 13/30 for a 90-degree shaft angle. Tables give general dimensions; tooth proportions for 1-diametral pitch; recommended tolerances for outside diameter, crown to back, and face angle; and fine-pitch straight bevel dimensions. Sketches show the important bevel gear blank dimensions and the application of tolerances to bevel gear blanks. Tooth proportions given are similar to those of the AGMA Standard on Straight and Spiral Bevel Gears with the following modifications: Clearances are increased; tooth thicknesses correspond to those generated by a crown gear in which the tooth thickness and space width are equal; the maximum face width is limited to three-tenths of the cone distance, or 8 divided by the diametral pitch, whichever is smaller. Copies of this standard may be obtained from The American Society of Mechanical Engineers, 29 West 39 St., New York 18, N. Y., for $1.00 each.

20-Degree Involute Fine-Pitch System for Spur and Helical Gears ASA B6.7-1950: Spur gear portion of American Standard closely follows the 1932 Spur Gear Tooth Form Standard with a slight increase in the whole depth to allow for greater proportional

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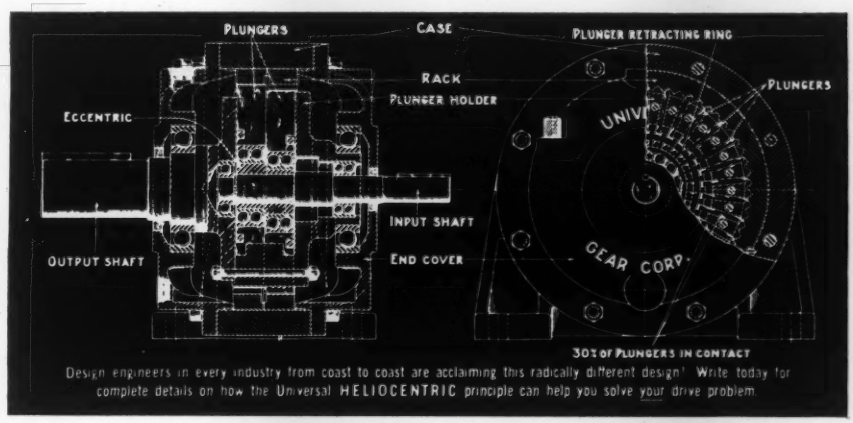

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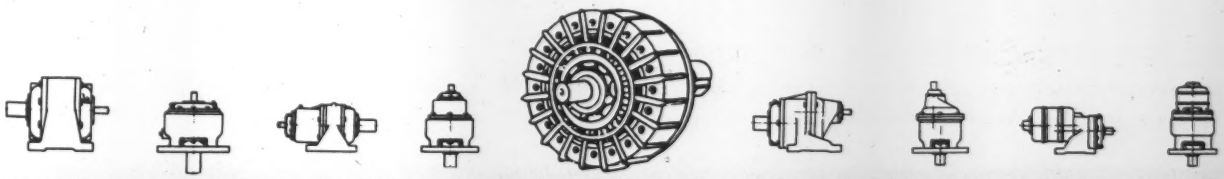

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Manufacturer & Association Publications

Laminated Plastics Data for Industry: A comprehensive 96-page handbook for the design and production engineer. Properties and performance characteristics of every grade of Formica are tabulated. Included are sections on molding, postforming and fabricating laminated plastics. The handbook is bound in a durable loose-leaf binder to provide for subsequent additions. Copies may be obtained by sending a company-letterhead request and $1.00 in cash to The Formica Co., 4545 Spring Grove Ave., Cincinnati 32, O.

Professional Viewpoints

(*Concluded from Page* 184)

cause the "best" diameter determined by calculus will generally fall between two commercial wire sizes which must then be tried.

A good first approximation to the "best" spring can be obtained by considering that the spring of minimum weight will result if the total absorbed energy is a minimum, which will be the case if the solid load is twice the fixed given center load, $F_e = 2F = 2U/f$. This relation can be proved by calculus or by elementary considerations of symmetry. The rate, $k = 2F/(f+2b)$, of the lightest spring is thus given as well as the solid load $2F$, and a spring can be calculated from these. The shortest possible spring is generally somewhat stiffer, made of thicker wire than the lightest possible spring because more steel can then be packed into a given length. In our example it so happens that the shortest and the lightest spring for 133,000 psi solid stress are one and the same because the next larger wire size is too thick.

This property of springs, that least spring weight results when the solid load is twice the load "fixed" by the requirements, shows on inspection that the author's spring, with 225-lb solid load and 150-lb fixed center load cannot be the shortest spring, because it is more flexible than the lightest spring.

—H. O. Fuchs and R. I. Gardner
Preco Inc.
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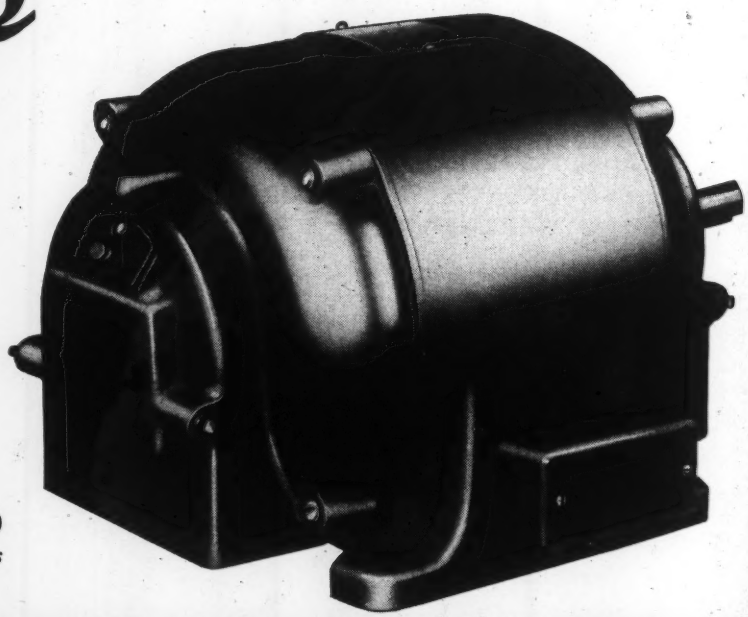

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MOLDS FOR PLASTICS

It sounds foolish to say that molds are what you make them, but they are, and can be a definite headache for that reason. When a customer buys the cheapest mold that is offered to him, he may easily be heading for trouble. That same purchasing agent would throw up his hands in horror if some one proposed that he buy the cheapest available truck to tote his goods around. He knows well that there are other things to trucks than price. He would say that upkeep and continuity of service were important too, and that he was willing to pay for the insurance of these two items in the original cost.

Because the mold is in some one else's shop, and that some one else is assuming the responsibility for these items, is not enough to justify ignoring the quality of the mold to be purchased. Regardless of the blame being placed elsewhere for a breakdown in production, the purchasing agent will be the one to hear from the front office.

The most talented molder in the world cannot make a bad mold stand up to a heavy schedule. In fact, 90% of molding troubles, from accuracy to ultimate delivery, are traceable to inferior molds or mold design. And most of these in turn trace back to pressure from some ambitious client who decided he would save money on the molds after the piece price had been settled. Add to that, of course, a weak, hungry molder who yielded in the hope that this time he could get away without trouble. Some time he does.

George K. Scribner, President
BOONTON MOLDING CO., BOONTON, N. J.
Phone: BOonton 8-2020

NEWS OF MANUFACTURERS

The Chemical Department of the **General Electric Co.** has discontinued the fabrication of laminated plastics at the Coshocton, O., plant of its Laminated and Insulating Products division. This operation, which produced fabricated parts from Textolite laminates, has been sold to the **General Laminated Products Inc.** with plants in New York and Chicago. General Electric will continue to produce Textolite laminates in the form of sheets, tubes and rods at Coshocton. It will also continue to manufacture decorative laminated plastics surfacing materials and custom molded laminated products.

♦

Chicago Belting Co., a manufacturer of leather and synthetic rubber products, has formed a new company, a wholly owned subsidiary, to be known as the **Allis Seal Corp.** Creation of the new company was prompted by Chicago Belting's interest in expanding its mechanical seal service.

♦

The George L. Nankervis Co., Detroit, Mich., manufacturer of plating, buffing and metal-finishing equipment, has purchased the **Electro-Mechanical Devices Co.** of Detroit, as a wholly owned subsidiary. Electro-Mechanical (EMDCO) designs and builds dynamometers, control panels, test stands and special machinery for the automotive and aircraft industries. The subsidiary company will be known as the **Electro-Mechanical Devices Division of The George L. Nankervis Co.** The two companies are consolidating their facilities in a new plant located at 19255 West Davison, Detroit, Mich.

♦

Fairchild Engine Division is the new name for the Ranger Engines Division of **Fairchild Engine and Airplane Corp.** Located at Farmingdale, Long Island, the division is engaged in the development of various types of jet engines.

♦

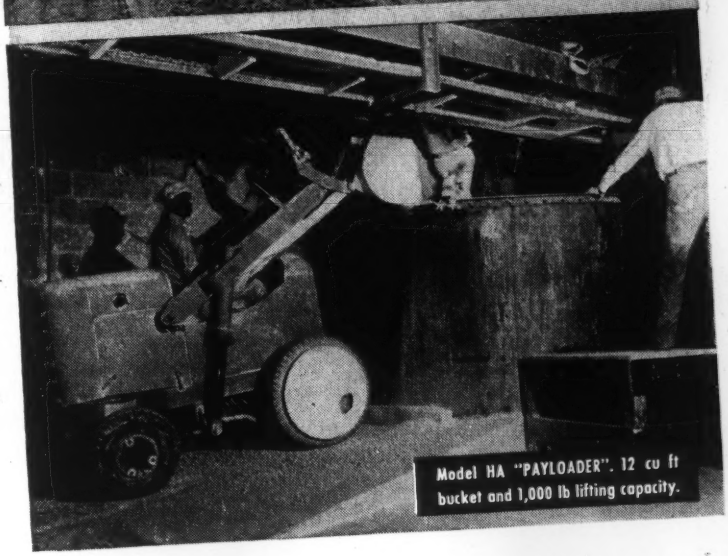
Lebanon Steel Foundry, Lebanon, Pa., has effected licensing agreements with **The Timken Roller Bearing Co.** to produce and sell a heat-resistant alloy, "16-25-6". Patent and trademark registrations are owned by Timken. Adaptation of the material to steel castings is underway to broaden Lebanon's production of jet engine components and other heat-resistant steel castings of aircraft quality.

♦

The **S. H. Couch Co.,** North Quincy, Mass., manufacturer of communication and signaling equipment, has purchased the Anthony pushbutton formerly manufactured by the **Anthony Machine and Electrical**

VICKERS HYDRAULICS *Help* HOUGH "PAYLOADERS" *LIVE UP TO THEIR NAME*

Model HM "PAYLOADER". 1½ cu yd bucket and 6,000 lb lifting capacity.



Model HA "PAYLOADER". 12 cu ft bucket and 1,000 lb lifting capacity.

Vickers Hydraulics help various models of the Hough "PAYLOADER" live up to their name . . . a name that means fast, low cost operation. Typical are Models HM and HA shown here; a Vickers Series V-200 Pump supplies hydraulic power. The big Model HM also has Vickers Hydraulic Power Steering.

The Vickers hydraulic equipment used on the Hough "PAYLOADERS" is representative of only one series of units built expressly for mobile and construction machinery service. For example, the new Vickers Balanced Vane Hydraulic Pumps, Series V-200, V-300 and V-400 offer the designers and manufacturers of this class of machinery a wide range of pump capacities . . . capacities of 2 to 54 gpm for hydraulic operating pressures up to 1500 psi.

Vickers Hydraulics have improved the performance of a wide variety of mobile equipment. It will pay you to look into the possibilities of using the best hydraulic pump value on your equipment. Get in touch with the Vickers office nearest you or write for further information.

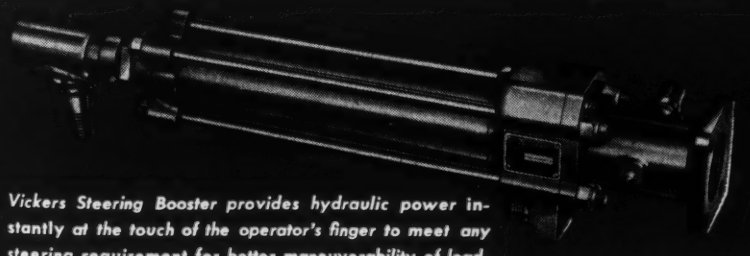
VICKERS Incorporated
DIVISION OF THE SPERRY CORPORATION
1430 OAKMAN BLVD. • DETROIT 32, MICH.

Application Engineering Offices: ATLANTA • CHICAGO • CINCINNATI CLEVELAND • DETROIT • HOUSTON • LOS ANGELES (Metropolitan) • MILWAUKEE NEW YORK (Metropolitan) • PHILADELPHIA • PITTSBURGH • ROCHESTER ROCKFORD • ST. LOUIS • SEATTLE • TULSA • WASHINGTON • WORCESTER

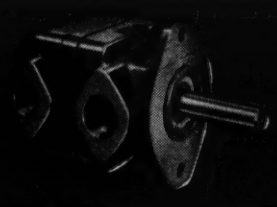
ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921

4293

Vickers Hydraulic Power Steering is Effortless, Positive and Shockless



Vickers Steering Booster provides hydraulic power instantly at the touch of the operator's finger to meet any steering requirement for better maneuverability of loaders with heavy loads on steering axle.



This Series of Balanced Vane Type Pumps features automatically controlled radial and axial clearances that maintain high efficiency over a very long life.

THIS 3¼ INCH DIAMETER CIRCLE SHOWS ACTUAL SIZE CROSS SECTION OF THE 3" BORE O-M CYLINDER . . . .

PACK POWER IN ⅓ LESS SPACE

O-M CYLINDERS SAVE SPACE PLUS MAINTENANCE AND ORIGINAL COST

● O-M hydraulic, water, air cylinders save up to ⅓ in space required because tie rods and bulky end caps are eliminated. Unique circumferential key locks honed steel tubing to compact end caps. That means *more* power in the space required by conventional cylinders . . . or the same power in less space.

Maintenance is simplified with disassembly and repacking only a matter of minutes. Ports may be oriented radially to an infinite number of positions. A full line of mounting brackets is available . . . interchangeable (bore for bore) on all O-M cylinders. That means lower first cost, smaller inventory.

Bore sizes to 8" immediately available. Steel (no castings) and bearing bronze throughout. Write today for specifications.

ORTMAN-MILLER
MACHINE CO., INC.
1210 150th Street, Hammond, Indiana

Works Co., Boston, Mass. Production of the new product, which can be furnished in several variations including multiple button blocks, has begun at Couch's North Quincy plant.

♦

The Kirk & Blum Mfg. Co., Cincinnati, O., manufacturer of dust and fume control systems, industrial drying and baking ovens, sheet metal parts and assemblies, weldments, and electrical control boards and enclosures, has acquired the entire plant and property of the **Cincinnati Planer Co.** The plant, located at 3120 Forrer St. in suburban Oakley, will be occupied by Kirk & Blum toward the end of the year.

♦

Packard Motor Car Co., Detroit, Mich., has embarked on a $1,000,000 plant improvement program to be completed by the end of the year. Topping the list of expenditures will be the erection of four test cells and a subzero laboratory room at the Packard proving grounds, and the installation of new equipment in the powerhouse.

♦

The **United States Rubber Co.** has begun work to reactivate the GR-S synthetic rubber plant at Port Neches, Tex., as requested by the Reconstruction Finance Corp. It is expected that the plant will be in operation in about three months. One of the first jobs will be to reassemble equipment which was dismantled and put into standby when the plant was closed after World War II. The Port Neches plant has a rated capacity of 60,000 tons with the possibility of increase to 75,000 tons if more production is needed. United States Rubber is operating a Government-owned synthetic plant at Naugatuck, Conn., with a capacity of 30,000 tons, and also a plant at Borger, Tex., with a capacity of 45,000 tons. The company will continue to operate the Borger plant until Port Neches is fully reactivated, after which operation of Borger will be turned over to the **Phillips Chemical Co.**

♦

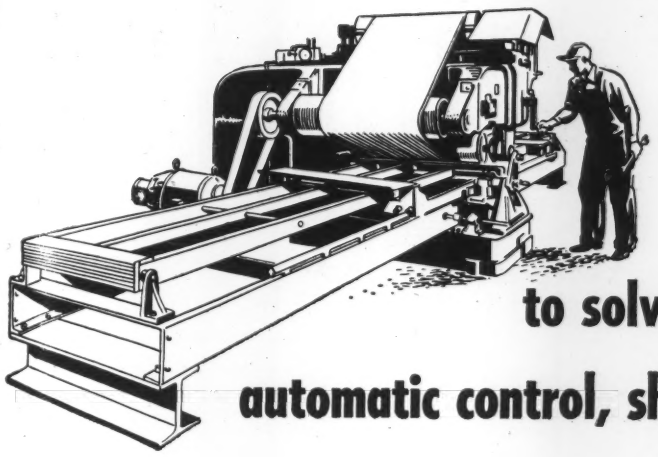
Pitney-Bowes Inc., manufacturer of postage meters and mailing machines, is starting a $250,000 building expansion program that will add 36,000 sq ft of floor space to its plant and general offices at Stamford, Conn. This marks the company's second expansion in five years. The new construction will add an entire fourth floor to the administration building and extend the west wall of the plant to enlarge machine shop and assembly operations.

♦

Universal joints, as manufactured in the United States by the **Mechanics Universal Joint Division** of **Borg-Warner Corp.,** will be produced soon by **Repco Ltd.,** Melbourne, Australia. A new division of Repco—**Repco Mechanics Proprietary**—will be created and a new plant will be erected or leased in Melbourne.

♦

Dudek and Bock Co. is planning the addition of two floors to its present 8,000-sq ft plant, 2100 West Fulton St., Chicago, Ill. Added facilities will provide means of handling the increased demand for coil springs, metal stampings and wire forms.



Smart design engineers use WARNER ICB UNITS to solve problems of clutching, braking, automatic control, shock and torque on their products

●IF YOUR PROBLEM

is to design your machine or its activating parts so that *you get:* (1) Push-button actuation at the machine or a remote control panel. (2) Fully or semi-automatic operation in "stop" or "start" of machine cycles. (3) Controlled acceleration or deceleration. (4) Automatic or manual adjustment of torque to load . . . or if your problem is *to eliminate:* (1) "Shock" from stop or start on gear trains and shafts. (2) Complex mechanical lever linkage, or air and hydraulic piping systems. (3) Vibration and hammering from solenoid operated brakes and clutches . . . or if you want to *increase production* from your machine by increasing the speed and number of cycle operations —

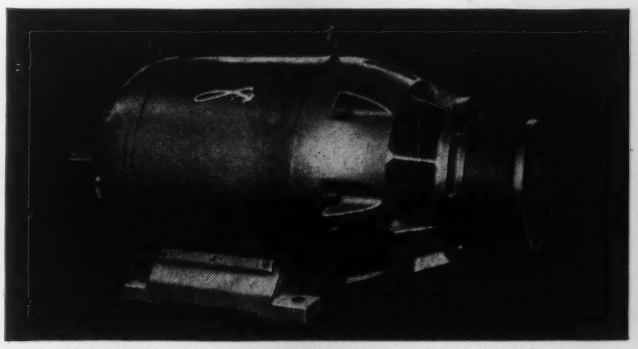
●A GOOD SOLUTION

to any one or combination of these problems is WARNER ICB Units. WARNER ICB Units are electrically controlled and operated clutches and brakes. They are compact, mechanically simple and may be applied to machine spindles, drive shafts, or direct to electric motors. Because of their design, they may be operated individually or in combination, manually or automatically, by direct or remote control. They provide for infinite controlled variation of torque in either clutching or braking action within their rated capacities. Their action is smooth and shockless—fast and practically noiseless. They are self-adjusting and self-compensating for life. Their design guarantees a high rate of heat dissipation.

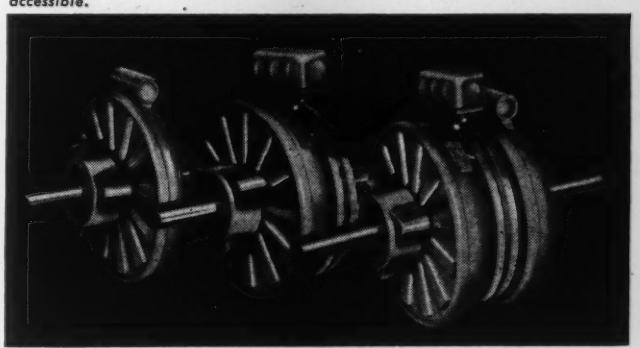
●MORE INFORMATION

on WARNER ICB Units and their application as clutches or brakes to various kinds of machinery is available. They may be used on continuous or intermittently operated machines. There is a complete range of sizes to meet both braking and clutching torque needs. If you have a design problem such as indicated above, write today for our free bulletin No. 701. If possible, state your problem. Not only will you receive your bulletin copy, but experienced WARNER ICB engineers will be glad to consult with you without obligation. Write direct to WARNER ELECTRIC BRAKE MANUFACTURING COMPANY, Beloit, Wis.

Bulletin 701 contains complete specifications and description of WARNER ICB Units. See for yourself how they offer advantages never before available on industrial clutches and brakes. Write for it today — no charge.



A WARNER ICB 9" x 2½" Brake Unit installed and housed on a standard 40 hp motor for a flat belt sander. Installation speeded up machine cycling, gave better, faster operational control. Installation is simple, compact and readily accessible.



From left to right: a Warner ICB Brake Unit, a Warner ICB Clutch Unit, and a combination Warner ICB Clutch and Brake Unit. Note clean, smooth, compact, appearance — simple mechanical parts.

WARNER ICB UNIT

WARNER ICB Units are manufactured by Warner Electric Brake Mfg. Co. — pioneers in the field of electric brake design and application since 1927.

HERE are some of the many
"COMPO" AND "POWDIRON"
Structural Parts produced by BOUND BROOK

- they're all in production—not samples
- all physical properties actually meet Bound Brook claims
- dimensional tolerances are held
- and, there's no mystery, black magic, or blue sky painting in our approach to parts

"COMPO"—Porous Bronze
"POWDIRON"—Sintered Iron
also Copper, Brass and many other alloys

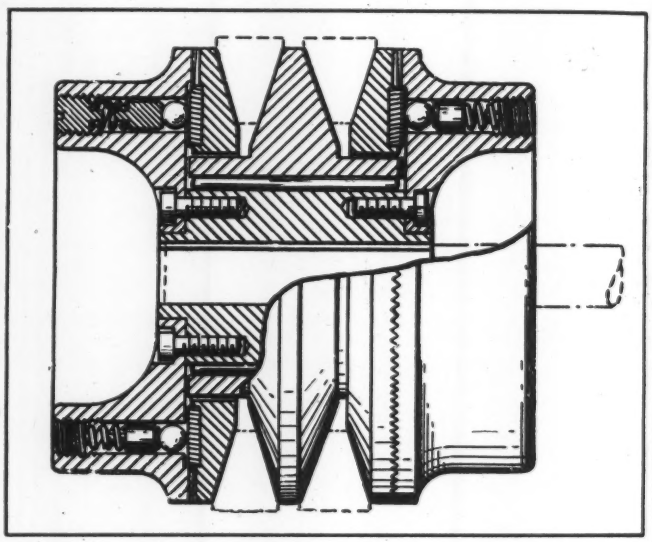
CONSULT OUR ENGINEERS—
they've solved many
a structural part problem

BOUND BROOK

"COMPO" • "POWDIRON" • BOUND BROOK

NOTEWORTHY PATENTS

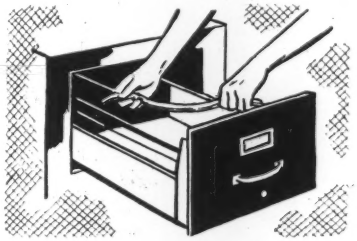
AUTOMATIC ENGAGEMENT of a driven shaft with its driving sheave is controlled only by tension of the V-belt drive described in patent 2,491,464, granted to Lysle L. Hazen and assigned to Enterprise Machine Parts Corp. Tightening the belt or belts causes the sheave sections to move apart against spring pressure, engaging jaws on a clutch plate attached to the input shaft. When belt tension is decreased, the jaws disengage, permitting the input shaft to run free of the sheave. Slip at a predetermined torque can be obtained by substituting friction surfaces for the jaws between sheave and clutch plate.



CONSTANT CLEARANCE is maintained between the drum and friction surfaces of a hydraulic brake mechanism as described in patent 2,493,250, assigned to United States Rubber Co. by Henry H. Kerr. A tension coil spring tends to retract the brake shoes radially while U-shaped spring clips on the shoes oppose movement of the shoes. Applying hydraulic pressure moves the shoes outward against spring tension and against the static frictional force of the friction clips.

Releasing the hydraulic pressure permits the spring to retract the shoes until they contact the clips, a distance equal to the desired shoe clearance. Since frictional force of the clips is greater than the radial effect of the spring tension, the shoes can then contract no further. As the lining wears, the position of the

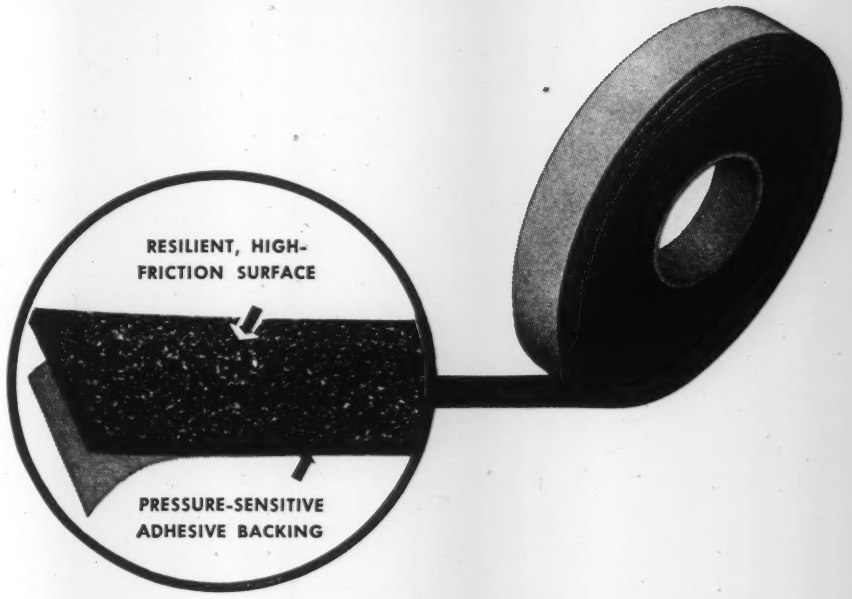
WIRES WON'T CHAFE as they pass through aircraft bulkheads when padded with Armstrong's DK-153.



FOLDERS WON'T SLIDE down out of sight in file drawers that have been stripped with Armstrong's DK-153.

QUIETER BUSES are made by applying anti-squeak DK-153 Tape between metal body panels and frame.

VIDEO TUBES ARE PROTECTED from damage during shipment and installation with Armstrong's DK-153.



A quick way to stop squeaks and rattles

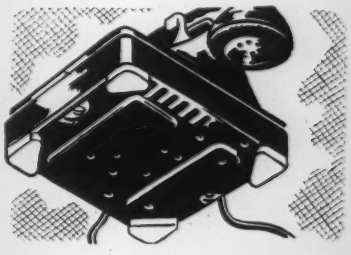
Squeaks and rattles disappear when you strip Armstrong's DK-153 Tape onto potential trouble spots. DK-153 Tape puts a resilient cushion of cork-and-sponged-rubber between adjoining surfaces. It stops squeak-producing slippage and absorbs vibration and rattles. Its pressure-sensitive adhesive sticks to any clean, dry surface.

High surface friction combined with resilience and ease of application makes DK-153 also ideal for a variety of other jobs. It cushions fragile parts like glass video tubes; it keeps easily marred enamel surfaces from being scratched during shipment; it makes office equipment non-skid on polished desk tops. Normally supplied in convenient rolls, Armstrong's DK-153 Tape is available also in shapes die-cut to your specifications.

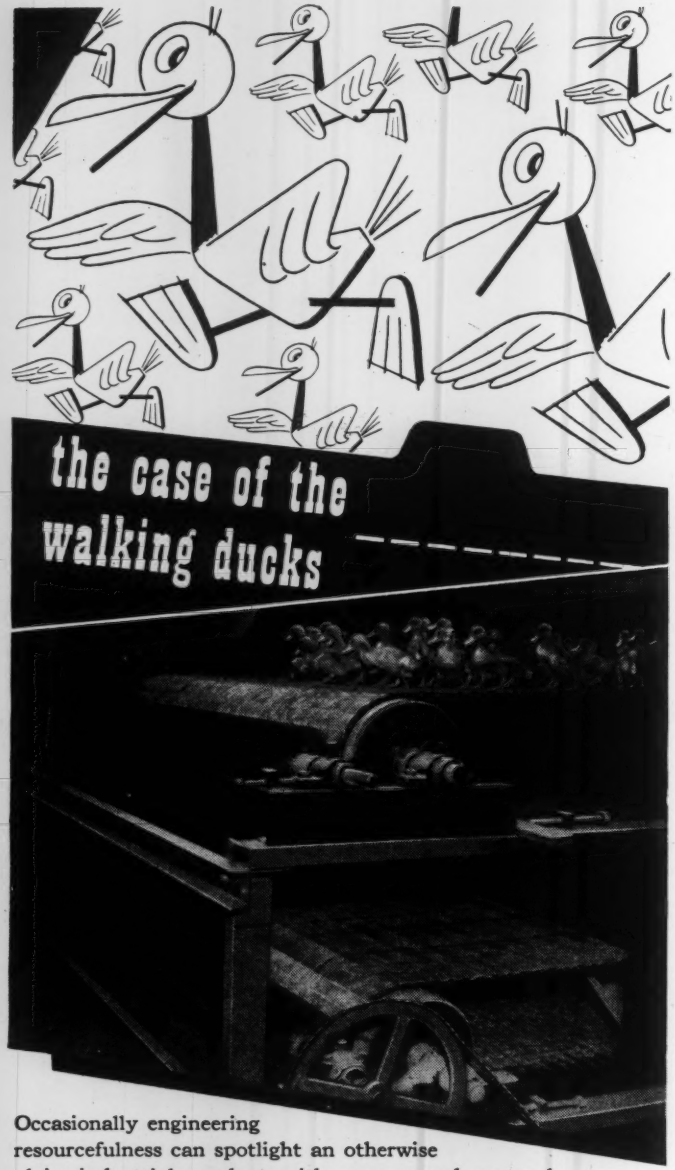
We'll be glad to send you free samples and further information. Just write Armstrong Cork Company, Gaskets and Packings Dept., 5108 Arch Street, Lancaster, Pennsylvania. Available for export.

SQUEAKS ARE ELIMINATED in diesel locomotives and railway coaches with DK-153 Tape on rubbing parts.

ASSEMBLIES ARE SPEEDED by using die-cut pieces of Armstrong's DK-153 Tape as anti-mar lamp base pads.



DIALING IS EASIER on polished desks and counters when telephones are equipped with non-skid DK-153.



Occasionally engineering resourcefulness can spotlight an otherwise plain industrial product with a new and unusual use . . . such as the case of this Woven Wire Conveyor Belt installation in a duck brooder.

Newly hatched ducklings, it seems, have to be kept moving to be kept alive. Placed on this specially built woven wire belt moving at a slow rate of speed, the ducklings remain there for eight days . . . continually treadmilling away from the falling end of the belt. Mortality rate dropped to zero. By-product recovery . . . duck droppings for fertilizer . . . is aided by the open mesh of the woven wire belt.

Even if you don't operate a duck hatchery, call in a Cambridge engineer whenever you have a problem concerning the movement of materials or products during processing. Let him explain the superior construction features of . . .

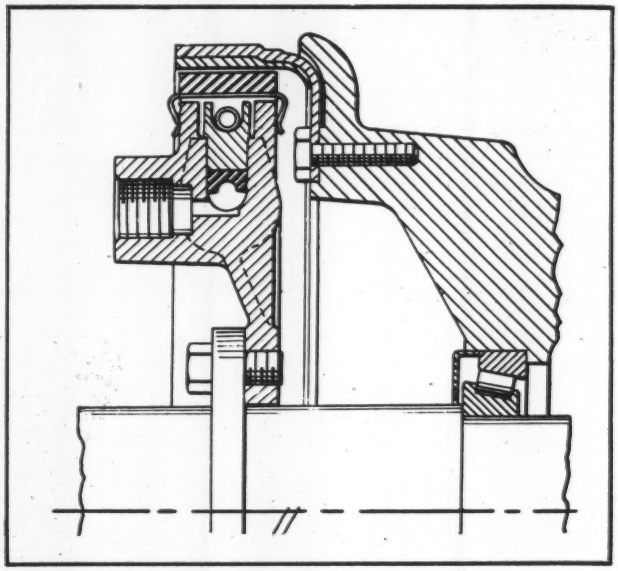
CAMBRIDGE Woven Wire Conveyor Belts

This illustrated 130 page catalog—FREE—to men concerned with production control. WRITE TODAY.

Cambridge Wire Cloth Co.

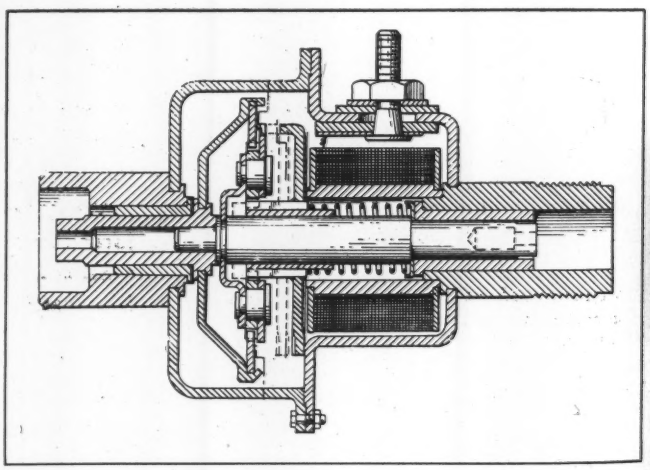
Department N • Cambridge 8, Maryland

NEW YORK • BALTIMORE • PITTSBURGH • SAN FRANCISCO
BOSTON • DETROIT • CHICAGO • ST. LOUIS • HOUSTON



clips is automatically adjusted radially outward, as hydraulic pressure is applied, to maintain the clearance regardless of lining wear.

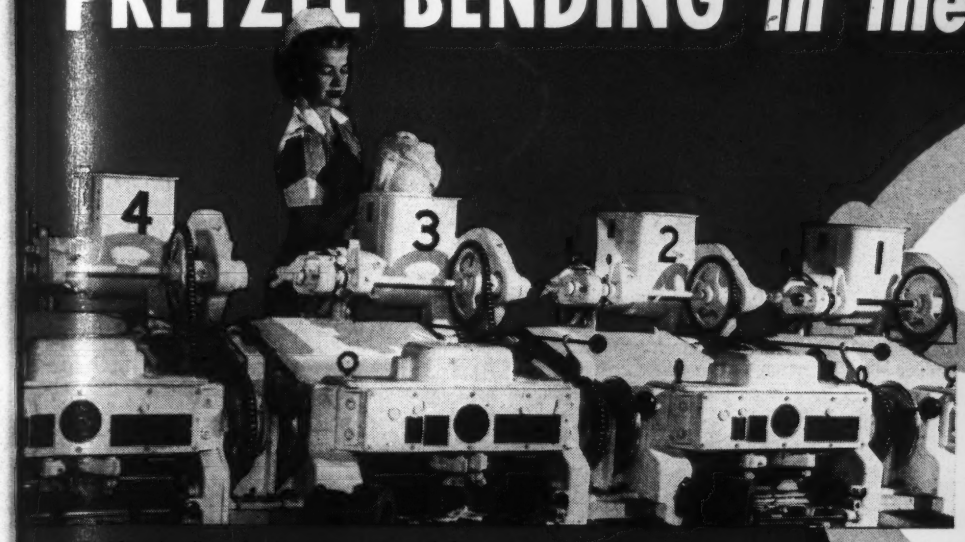
MAGNETIC CONTROL of planetary gearing provides two-speed operation of the transmission covered in patent 2,500,447. When the unit is driving at the one-to-one ratio, the coil is de-energized and a spring has moved a disk axially to lock the ring gear, on the input shaft, to sun and pinion gears, resulting in a direct drive through the mechanism. Actuating the magnetic coil unlocks the planetary train and holds the sun gear stationary through magnetic attraction between the coil and a plate which engages



the sun gear. In reduced speed drive, power is transmitted from the ring gear through the planetary pinions, which run around the sun gear, to the pinion cage on the output shaft. The patent has been assigned to Eaton Mfg. Co. by Fred Bitzer.

Complete printed copies of all patents are available from the Commissioner of Patents, Washington 25, D. C., for 25 cents each.

PRETZEL BENDING *in the New Manner*



ABOVE: After loading the hoppers and until formed pretzel is dropped on belt at the rate of about one every second, the operation is entirely automatic. Positions of the Diamond Roller Chains drives are the same in each machine.

BY DIAMOND Roller Chain Driven AMF Pretzel Machines

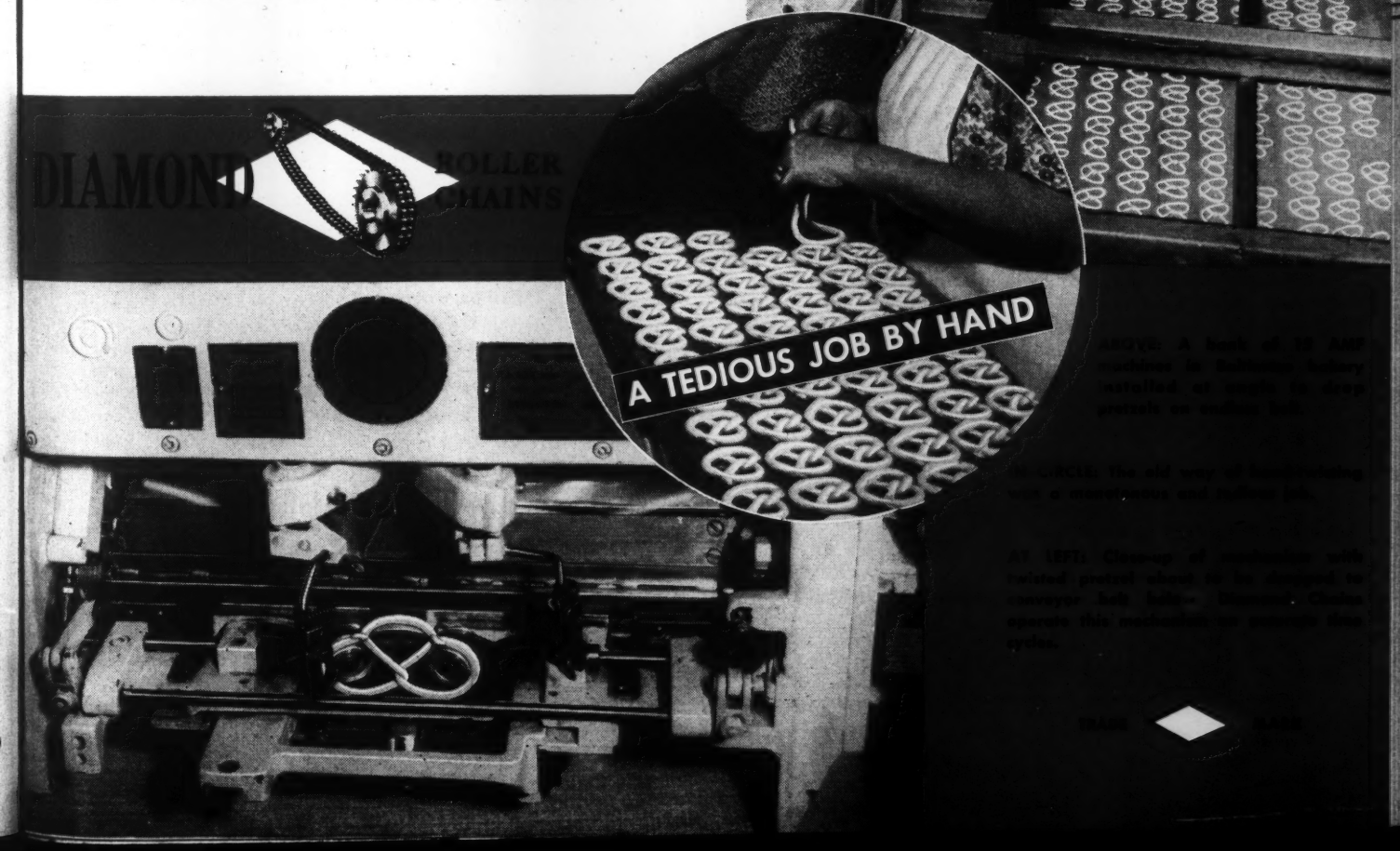
Correct Timing Relationship

• To produce pretzels of the favored variety, American Machine and Foundry Company have perfected a machine that does the work mechanically.

For this ingenious automatic machine, non-slip Diamond Roller Chain Drives are used from main line shaft to jack shafts and from these shafts to the rolling boards and extruding mechanism which includes dough cut-off knives and friction clutch that drives the mechanical fingers. Exact timing relationship between the various mechanisms of the machine is maintained.

Your machinery applications may be considerably different but we are sure Diamond Chains can serve you equally well. DIAMOND CHAIN COMPANY, Inc., Dept. 435, 402 Kentucky Avenue, Indianapolis 7, Indiana.

Offices and Distributors in all principal cities.



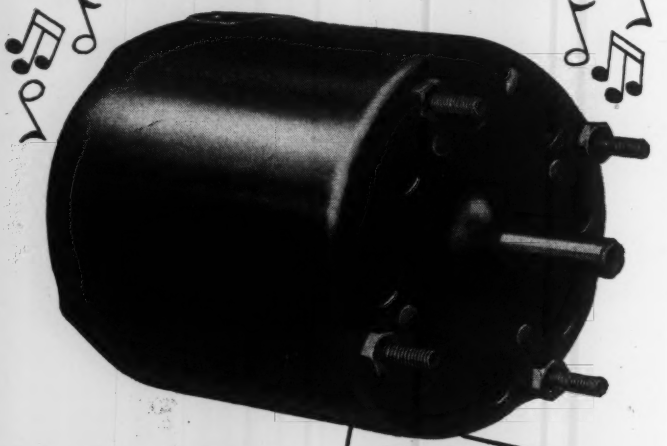
ABOVE: A bank of 12 AMF machines in Bohemian bakery installed at angle to drop pretzels on moving belt.

AT RIGHT: The old way of hand-twisting was a monotonous and tedious job.

AT LEFT: Closeup of mechanism with twisted pretzel about to be dropped to conveyor belt below. Diamond Chains operate this mechanism on separate drive cycles.

FASCO
FRACTIONAL H. P. MOTORS

. . . Help *Revere* Capture Sound Perfectly



REVERE Camera Company, maker of famous REVERE 8mm and 16mm movie equipment, features tone perfection and important mechanical advantages in the REVERE Magnetic Tape Recorder. A high quality FASCO motor assures *quiet* operation . . . eliminates sound distortions caused by variation in speed. Tape starts and stops quickly . . . no reel coasting, no loss or slur of sound . . . fast rewind and forward speeds save time. All *real* advantages . . . examples of efficient, practical FASCO motor performance.

There's a complete line of FASCO Fractional H.P. shaded pole motors from 1/500 to ⅛ H.P. to help *you* improve your product, or pack maximum efficiency and economy into the design of your new one.

"Your comparison test, too, will prove FASCO best"

WRITE for complete information on FASCO Fractional H.P. Motors.

FASCO
INDUSTRIES, INC.
ROCHESTER 2, NEW YORK

SOCIETY ACTIVITIES

The **American Society of Mechanical Engineers** will conduct its 1950 Fall Meeting at the Hotel Sheraton, Worcester, Mass., Sept. 19-21. The Machine Design division has scheduled two sessions to be held on Sept. 20. The following papers will be presented:

"A Main Spring Dynamometer," J. E. Hendricks, supervisor of the Watch Research Laboratory, Hamilton Watch Co., Lancaster, Pa.

"The Design of Nonlinear Springs," Stanley P. Clurman, project engineer, special weapons eng., Sperry Gyroscope Co., Great Neck, Long Island, N. Y.

"On the Design of Rotor Coil Support Rings," James J. Ryan, professor of mechanical engineering, University of Minnesota, Minneapolis, Minn.

"Styling the Machine Tool," Harold Sizer, assistant to director of design, and Frank Burgess, Brown & Sharpe Mfg. Co., Providence, R. I.

Several papers to be presented under the auspices of other ASME divisions will also be of interest to design engineers. Tentatively planned inspection trips include: General Electric Co., Crompton & Knowles Loom Works, Wyman-Gordon Co., Reed-Prentice Corp., Worcester Pressed Steel Co., Norton Co., Heald Machine Co., Bay State Abrasive Products Co., Telechron Inc., General Motors Corp., and Heywood-Wakefield Furniture Plant.

♦

In recognition of their eminence in technical work in engineering materials and for especially meritorious service to the organization over many years, the **American Society for Testing Materials** signally honored six of its members by awarding them Certificates of Honorary Membership at its Annual Meeting in Atlantic City:

Phaon H. Bates, retired, St. Petersburg, Fla.
Wilson C. Hanna, chief chemist and chemical engineer, California Portland Cement Co.
Dean Harvey, consultant on materials, Pittsburgh, Pa.
Prevost Hubbard, retired, New York City, N. Y.
Robert Job, retired, Montreal, Canada
Henry S. Rawdon, retired, Washington, D. C.

The men honored have had distinguished careers in various fields, but have done particularly notable work in various phases of ASTM work involving standardization and research. Presentation was made by the Society's President, James G. Morrow.

♦

The Illinois Institute of Technology will be host to the first **National Congress of Applied Mechanics,** June 11-16, 1951, in Chicago. Plans have been completed to hold a National congress every four years. Papers for presentation at the congress must be submitted

BALL BEARINGS
ROLLER BEARINGS
THRUST BEARINGS AND
BALL BEARING HOUSINGS

Single Row

Pillow Block

Engineered to the Bearing Industry's Highest Standards of Uniformity, Quality and Performance

Double Row

Take-up Unit

Self-Aligning

Flange Unit

AHLBERG BEARING COMPANY

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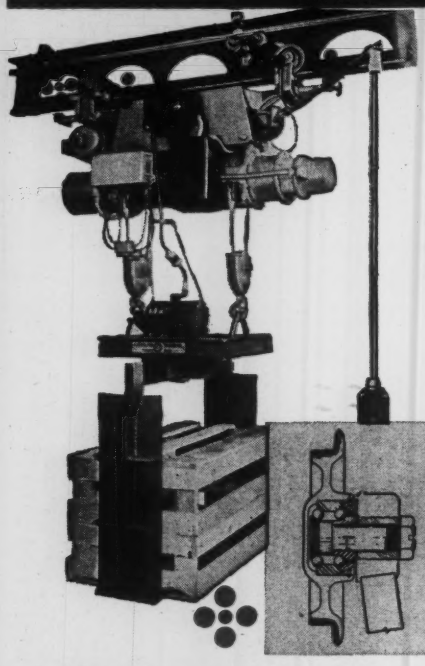
PRECISION CRAFTSMEN SINCE 1908



Over 30,000 installations . . . handling nearly every conceivable type of material in practically every industry —that's the agressive materials handling record behind Cleveland Tramrail systems.

Aetna double race type wheel bearing No. A-827 as used in Cleveland Tramrail Systems.

Included among their sturdy components are hard working wheel bearings to meet the high load capacity, precision smoothness and long, trouble-free performance required in this application—bearings which Aetna has proudly supplied for over 14 years.

Greater efficiency is an advantage which comes to any hard working equipment when Aetna bearings lend a hand. For money saving, problem solving counsel on your next bearing application, submit your prints to Aetna. No obligation, of course!

AETNA BALL AND ROLLER BEARING COMPANY
4600 SCHUBERT AVENUE • CHICAGO 39, ILLINOIS

Aetna

Standard and Special Ball Thrust Bearings • Angular Contact Ball Bearings • Special Roller Bearings • Ball Retainers • Hardened and Ground Washers • Sleeves • Bushings

before April 14, 1951, and should constitute original research in applied mechanics which includes kinematics, dynamics, vibrations, waves, mechanical properties of materials and failure; stress analysis elasticity, plasticity; fluid mechanics; and thermodynamics. Papers should not be longer than 5000 words. The following societies and universities are sponsoring the congress: American Institute of Chemical Engineers, American Society of Civil Engineers, American Society of Mechanical Engineers, American Mathematical Society, American Physical Society, Institute of Aeronautical Sciences, Society for Experimental Stress Analysis, U. S. National Committee on Theoretical and Applied Mechanics, Illinois Tech, Purdue University, Northwestern University, and the University of Illinois.

♦

At its Annual Meeting held recently at Hot Springs, Va., the **Malleable Founders' Society** elected the following officers:

President
Ralph T. Rycroft, vice president, Kencroft Malleable Co. Inc.
Vice President
Cal C. Chambers, president, Texas Foundries Inc.
Directors
W. V. Tiscornia, executive vice president, Auto Specialties Mfg. Co.
Max Traum, president, Newark Malleable Iron Works
C. M. Lewis, president, Badger Malleable & Mfg. Co.
Cal C. Chambers, president, Texas Foundries Inc.

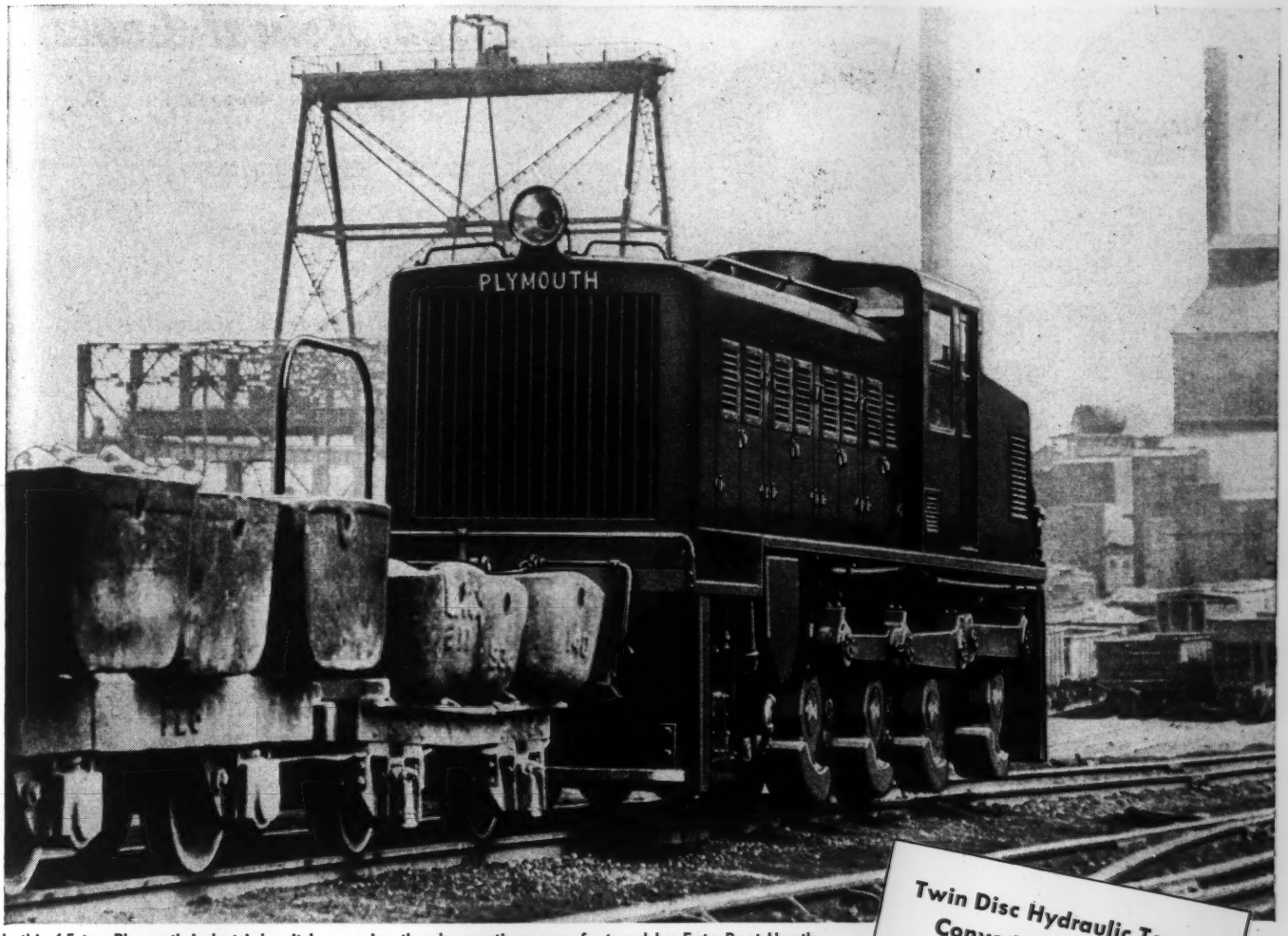
At the meeting of the directors, Lowell D. Ryan of Cleveland was renamed managing director and secretary and James H. Lansing of Cleveland was appointed Technical and Research Director again.

♦

J. Calvin Brown, owner of the Los Angeles firm bearing his name, has been nominated as the 1951 president of **The American Society of Mechanical Engineers.** Mr. Brown heads a slate of new ASME officers including four regional vice presidents and two directors-at-large, submitted by the nominating committee. Since only one name is presented for each office, nomination is tantamount to election. The entire slate shows:

President
J. Calvin Brown
Regional Vice Presidents
Henry Reginald Kesslar, manager, Republic Flow Meters Co.
Stephen Dewey Moxley, vice president, American Cast Iron Pipe Co.
Dr. John T. Rettaliata, dean of engineering, Illinois Institute of Technology
Carl J. Eckhardt, professor of mechanical engineering and superintendent of utilities, University of Texas
Directors-at-Large
Lionel J. Cucullu, assistant to chief engineer, New Orleans Public Service Inc.
Harold E. Martin, district manager, The Babcock & Wilcox Co.

Formal election will take place in the fall by letter ballot of the membership.



In this 65-ton Plymouth industrial switcher, as in other locomotives manufactured by Fate-Root-Heath, in the 12 to 70-ton class, you'll find Twin Disc Hydraulic Torque Converters . . . fluid power to eliminate rough starts, to handle more cars per day, and handle them cheaper.

Twin Disc Hydraulic Torque Converter Advantages

1. All shocks and vibrations absorbed. 2. Prime mover can't be lugged down or stalled. 3. Engine torque multiplied five times at stall. 4. Infinitely variable reduction ratio automatically adjusts to load conditions.

No Rough Starts Here

S-m-o-o-t-h-l-y . . . s-m-o-o-t-h-l-y . . . that's the way powered equipment picks up under full load when the engine delivers its power through a Twin Disc Hydraulic Torque Converter. There's no gear-shift guesswork: the Torque Converter offers an infinitely variable number of gear ratios *automatically.* Under starting or stall conditions, it *multiplies* engine torque as much as five times to increase tractive, hoisting, or other effort.

On this 65-ton Flexomotive Plymouth industrial locomotive a 16,000 series Twin Disc Torque Converter was specified to assure smooth, uninterrupted acceleration—output torque automatically adjusted to load requirements—eliminating the use of heavier, more expensive diesel-electric units. A planetary transmission with high, low and reverse is coupled with the Torque Converter to comprise the entire transmission.

This smoothness, this automatic selection of the right gear ratios, means more work done, longer equipment life. That's why more and more equipment manufacturers are switching to Twin Disc Hydraulic Torque Converters (Lysholm-Smith type)—available in 40 to 1,000 hp capacities. Write today for Bulletin No. 135-C.

TWIN DISC
CLUTCHES AND HYDRAULIC DRIVES

Clutches & Hydraulic Drives

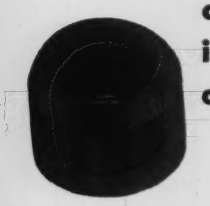
TWIN DISC CLUTCH COMPANY, Racine, Wisconsin • HYDRAULIC DIVISION, Rockford, Illinois

BRANCHES: CLEVELAND • DALLAS • DETROIT • LOS ANGELES • NEWARK • NEW ORLEANS • SEATTLE • TULSA

Purebon

solves many problems

caused by sliding or rotating parts which are difficult or impossible to lubricate



A constantly increasing stream of problems are facing engineers and designers today involving sliding or rotating parts where lubrication is difficult or impossible. For such applications, Purebon, the mechanical carbon, is often the ideal answer. Typical applications are seal rings, bearings, pistons, piston rings, pump vanes, valve seats, meter discs, and a host of similar items. Purebon comes in a wide variety of grades. It is strong, tough, readily machineable and in many cases can be molded directly to size.

Bulletin No. 482 tells the complete story of Purebon. Write for your copy today.

PURE CARBON CO., INC.
446 HALL AVE. ST. MARYS, PA.

Crossed Helical Gears

(*Continued from Page* 127)

whence $\psi = 59° 52'$. From Equation 7

$$d = \frac{2 \times 4}{1 + \sqrt{3.7}} = 2.74$$

and so $np_n = \pi \times 2.74 \times \cos 59° 52' = 4.32$

Selecting values of n from the combinations quoted above as closely approximating the desired ratio, it is found that $n = 14$ gives $p_n = 0.308$ which is close to 10 diametral pitch.

Accepting this and writing $p_n = \pi/10$ with $n = 14$ in Equation 8, it is found that $\cos \psi = 0.51$ or $\psi = 59° 20'$. This applies for $d = 2.74$, $n = 14$ and $N = 52$. Consequently, $\Sigma - \psi = 15° 40'$, sec $(\Sigma - \psi) = 1.039$, sec^3 $(\Sigma - \psi) = 1.12$ and p_n/π sec $(\Sigma - \psi) = 0.1039$. From Equation 2,

$$D = 0.1039 \left[51 + \frac{17 \cos^3 15° 40'}{51} \right] = 5.325$$

From Equation 3,

$$D = 0.1039 \left[54 - \frac{17 \cos^3 15° 40'}{54} \right] = 5.58$$

Now, $D = 2C - d = 5.26$, which lies below the permissible range. Hence d must be reduced by decreasing ψ. This will have the effect of raising the permissible range but the effect will be negligible as the change in sec $(75° - \psi)$ is much smaller than in sec ψ.

Tentatively it may be assumed that $D = 5.6$ will lie within the revised range. From this $d = 8 - 5.6 = 2.4$ and, from the left term of Equation 1

$$d = 2.4 = \frac{14}{\pi} \times \frac{\pi}{10} \sec \psi$$

or $\cos \psi = 0.583$ whence $\psi = 54° 20'$, $\Sigma - \psi = 20° 40'$, sec $(\Sigma - \psi) = 1.069$ and sec^3 $(\Sigma - \psi) = 1.22$. From Equation 2

$$D = 0.1069 \left[51 + \frac{17 \cos^3 20° 40'}{51} \right] = 5.48$$

From Equation 3

$$D = 0.1069 \left[54 - \frac{17 \cos^3 20° 40'}{54} \right] = 5.74$$

As $D = 5.6$ lies within these limits, these values may be accepted.

The equivalent numbers of teeth are 14 sec^3 54° 20′ = 69.3 and 52 sec^3 20° 40′ = 63.4. Therefore, $u = 63.4/69.3 = 0.91$

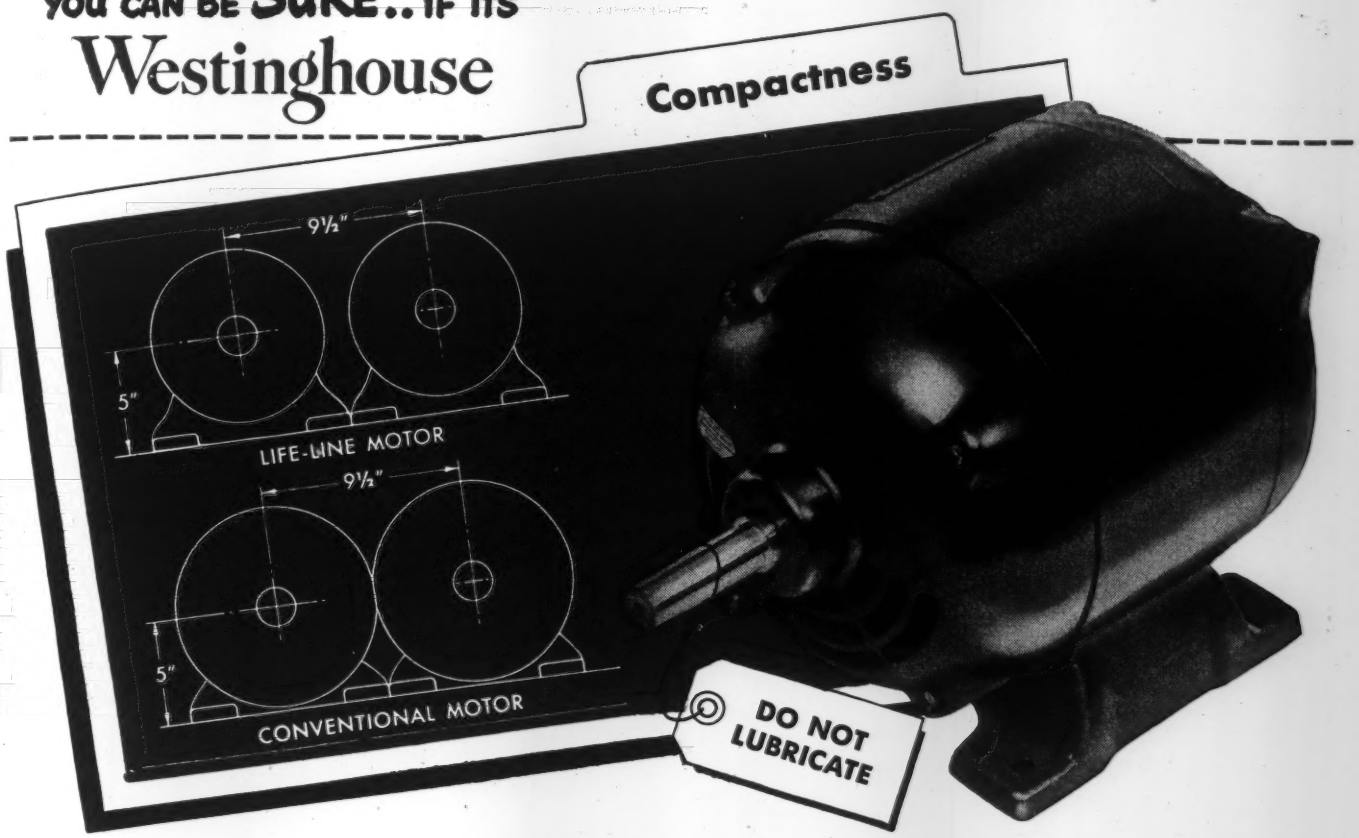
The 52-tooth gear is No. 1 and its blank diameter is 5.6 plus twice the addendum or

$$5.6 + \frac{2}{\pi} \frac{\pi}{10} (1.4 - 0.4 \times 0.91) = 5.807$$

The blank diameter of the 14-tooth gear is

YOU CAN BE **SURE**.. IF IT'S

Westinghouse



Life-Line's Space-miser compactness

SAVES PRODUCTION COSTS

This study in contrast shows how a Life-Line comes through in a pinch. For example: One manufacturer using conventional motors was forced to crowd two motors together. They did not quite fit, which meant removing a thin slice of cast iron from each motor frame. He'd been doing it for years . . . thought nothing of it.

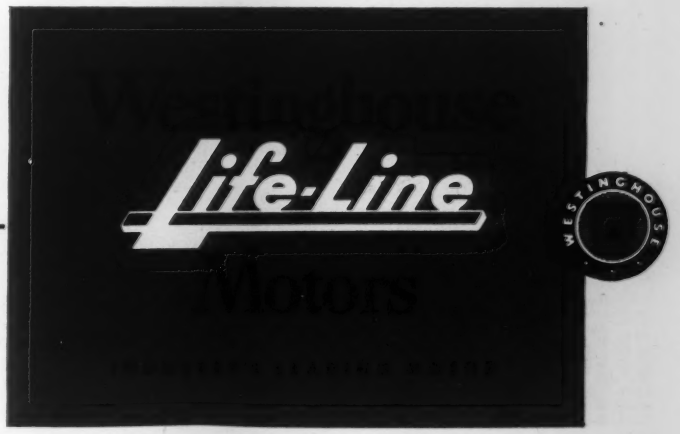
Then, a designer saw the all-steel Life-Line. He tried it. Result? A step saved in manufacturing. Life-Line's ⅓ smaller size, for the same frame size and hp rating, fit easily with room to spare.

This compactness is a valuable asset when your design allows only limited space for the motor drive. (And saves money in reduction of shipping weight, too.) In addition, Life-Line "needs no further lubrication". You can install it in "hard-to-get-at" locations—and forget it!

These are but a few of the pluses a designer has with the Life-Line. Ask your Westinghouse representative for others. Ask him for a copy of B-3482, or write Westinghouse Electric Corporation, P.O. Box 868, Pittsburgh 30, Pennsylvania.

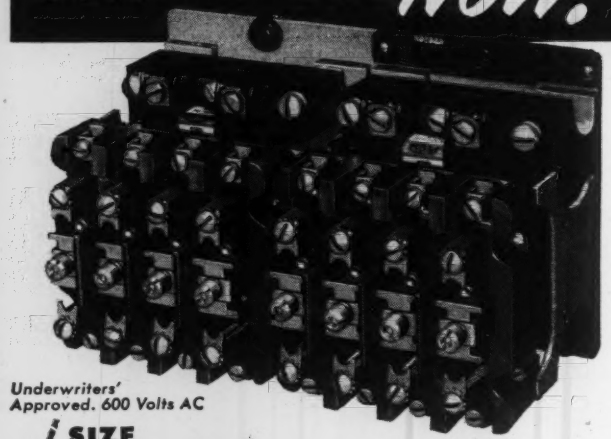
J-21575

— If you figure LIFE-COSTS . you'll figure LIFE-LINE —



Check

R-B-M INDUSTRIAL CONTACTORS NOW!



Underwriters' Approved. 600 Volts AC

SIZE

Non-Reversing

2 to 4 Pole 2-3/4" w. x 3-5/8" h. x 3-5/16" d.
5 to 8 Pole 5-9/16" w. x 3-5/8" h. x 3-5/16" d.

Reversing

2 to 4 Pole 5-9/16" w. x 3-5/8" h. x 3-5/16" d.

Note: 10 and 15 ampere contactors have same mounting and overall dimensions.

ACCESSIBILITY

To replace contacts, it is not necessary to disassemble the complete contactor. Just remove the parts comprising the stationary and movable contacts. Contacts can be replaced without disturbing wiring. To change coil, remove magnet frame and coil assembly only. (See illustration below.)

FLEXIBILITY

Using a screw driver only, you can easily change any pole from normally open to normally closed. No special parts required. 10 and 15 ampere parts are interchangeable.

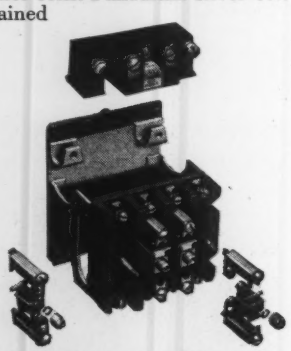
RELIABILITY

Laboratory tests involving millions of operations, plus field service of thousands of R-B-M contactors on door operators, radio transmitters, packaging and weighing machinery, hoists, machine tools and many other industrial applications offer proof of dependable, trouble-free performance.

ADVANCED DESIGN

Melamine Insulation. Molded coil housing. Ilsco solderless connectors. 50/60 cycle magnet coils. Palladium silver contacts. Stainless steel self-contained contact springs.

Where space is a factor, and accessibility a must—use R-B-M industrial contactors. Initial low cost plus dependable performance will save you money. Write for Bulletin 600 and price list on your company letterhead.



Dept. B-8, R-B-M DIVISION OF ESSEX WIRE CORP.

RBM

R-B-M DIVISION
ESSEX WIRE CORP.
Logansport, Indiana

MANUAL AND MAGNETIC ELECTRIC CONTROLS
—FOR AUTOMOTIVE, INDUSTRIAL, COMMUNICATION AND ELECTRONIC USE

$$2.4 + \frac{2}{\pi}\frac{\pi}{10}(0.6 + 0.4 \times 0.91) = 2.593$$

If the full depth corresponding to the cutter used is, say, 0.225-inch and if backlash of 0.010-inch is desired, the actual cutting depth should be $0.225 + 0.25 \times \cot 20° \times 0.010 = 0.232$-inch. Here

$$e = 4 - \frac{1}{20}[14 \sec 54° 20' + 52 \sec 20° 40']$$

$$= 4 - 3.97 = 0.03\text{-inch}$$

The addenda are 0.103 and 0.097-inch.

Minimum face width of pinion, f, is given by Equation 9 as

$$f = 2.5 \cot 20° \times \left[0.103 \sin 54° 20' + \frac{0.03 \sec 54° 20'}{\tan 54° 20' + \tan 20° 40'}\right]$$

$$= 6.875\left[0.084 + \frac{0.052}{1.77}\right] = 0.78\text{-inch}$$

while minimum face width of gear, F, is obtained as

$$F = 2.5 \cot 20° \left[0.103 \sin 20° 40' + \frac{0.03 \sec 20° 40'}{1.77}\right]$$

$$= 6.875\,[0.037 + 0.018] = 0.38\text{-inch}$$

CONTACT IN CROSSED HELICAL GEARS: In a pair of crossed helical gears, working at the center distance corresponding to Equation 1, (where $e = 0$) the points of contact always lie on two straight lines that intersect the common perpendicular to the shaft axes at a point that divides it in the ratio of the two terms on the left hand side of Equation 1. One straight line applies to one set of tooth flanks of each gear, and the other straight line to the other sets of flanks.

If the working center distance is such that e is not zero, then the two straight-line paths of contact do not intersect the common perpendicular to the shaft axes, but lie in planes equidistant from it and parallel to the normal plane. Because the contact-paths have moved in opposite directions from the common perpendicular, the width of each gear has to be increased above the minimum that suffices for the condition $e = 0$, to cover the two contact-paths.

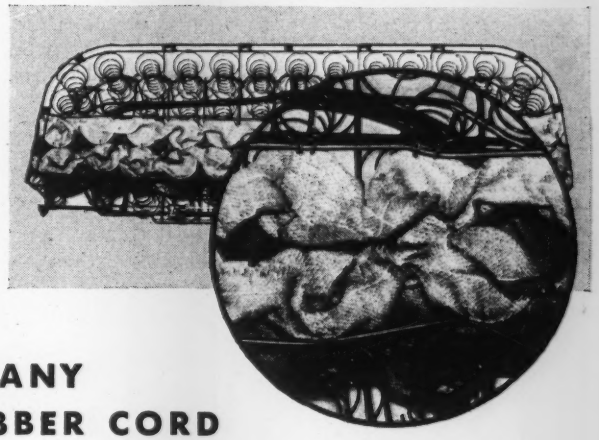
In spur or helical gears on parallel shafts, a condition in which e is not zero causes the teeth to have backlash if they are cut to standard full depth; in order to approximate the condition of no backlash the teeth must be cut to less than full depth.

This does *not* apply to crossed helical gears on non-parallel shafts. Cutting to standard full depth produces the same approximation to the no-backlash condition whatever the value of e. A variation of e from zero is not a reason for departing from full depth.

MINIMUM CENTER DISTANCE: If crossed helical gears are designed on the basis of Equation 1, with n, N and p_n fixed, no value of ψ can bring C below a certain minimum. This is determined in the case of $\Sigma = 90°$ by $\tan \psi = \sqrt[3]{(N/n)}$ but is not so simply expressed in other cases.

On the basis of the method described here, the minimum value of C for any given combination of n, N and p_n is lower than by conventional design methods.

How do you tie a Spring?



L. A. YOUNG SPRING & WIRE COMPANY USES STALWART-DEVELOPED RUBBER CORD

Confronted with the need for a tough, durable, elastic cord to tie seat springs in one of their many automotive cushions, L. A. Young Spring & Wire Company engineers specified Stalwart Rubber for help. Stalwart's chemists and engineers, despite the many difficult aspects to the problem, compounded a special rubber cord that incorporated all the desired characteristics.

To date, millions of feet of this cord have been delivered and installed in automobile seat cushions without the rejection of a single foot.

The development and production of this cord is another example of Stalwart's ability to supply top quality rubber parts to meet exacting requirements. Today, Stalwart is supplying custom parts to the automotive, aviation, refrigeration, and original equipment manufacturing industries. Thousands of molded, extruded, punched, lathe-cut and die-cut parts are fabricated daily from natural and synthetic rubber stocks (including Silicone rubber) to meet individual as well as S.A.E. specifications. WRITE FOR CATALOG TODAY, and . . .

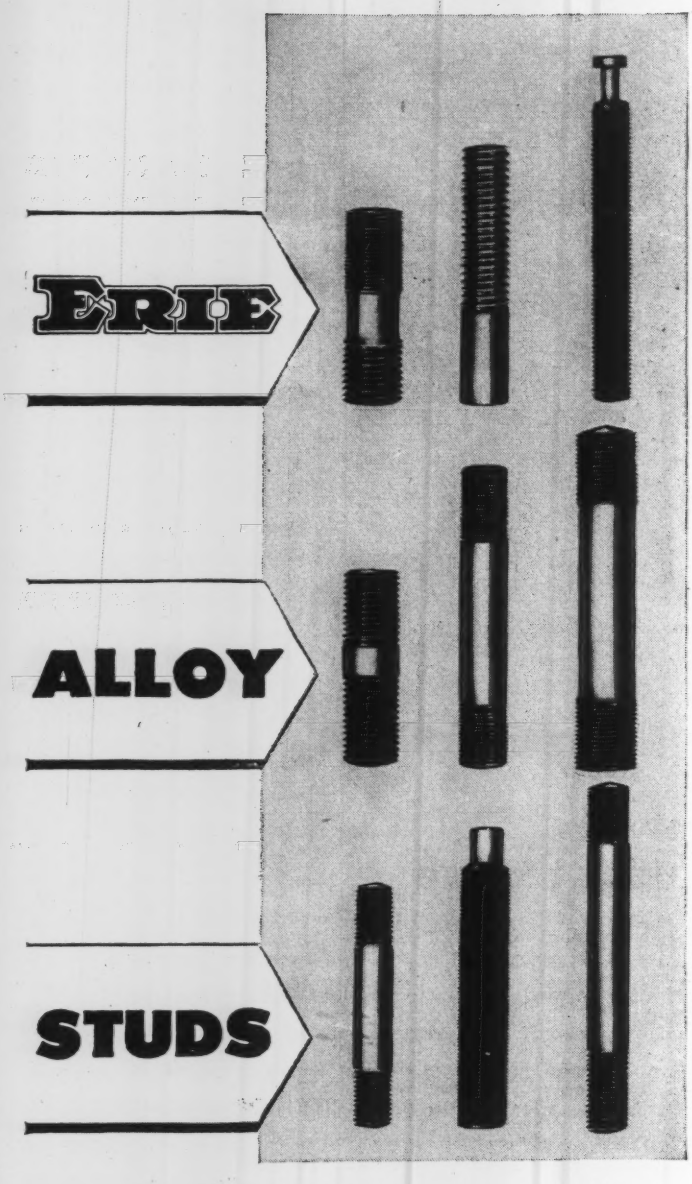
Specify STALWART . . . For Quality Custom Rubber Parts!

Stalwart Cord Meets These L. A. Young Specifications:

1. Tear Resistance . . . because of possible fracture during fabrication.
2. Chemical Resistance . . . because cord is installed prior to special paint dip.
3. Heat Resistance . . . to withstand oven paint drying process.
4. Flex Resistance . . . to withstand flexing of springs under passenger.
5. Age Resistance . . . because of extreme replacement difficulty if failure occurs prior to life expectancy of cushion.
6. Low Permanent Set . . . to maintain original length during processing operations and functional service life of seat.
7. Low Temperature Resistance . . . so that cord remains flexible at −40° F. to meet world-wide temperature conditions.
8. Abrasion Resistance . . . to withstand contact with metal parts.

STALWART RUBBER COMPANY

8180 NORTHFIELD ROAD • BEDFORD, OHIO



OUR 36 years of experience making high quality special bolts and studs, exclusively for thousands of users, is at your service. When your fastner demands depart from the standard line of stock studs in size, shape, materials, finish or type of threading please call on us for assistance.

Send your blueprints to ERIE for precision bolting.

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REPRESENTATION IN PRINCIPAL CITIES
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ALLOYS • STAINLESS • CARBON • BRONZE

Milestones In Design

(*Concluded from Page* 142)

impressed on the wire in undulations, so to speak, of magnetisation, a kind of writing which is permanent, and faithfully records the articulations of the voice. When *E* is put, now, in direct connection with a moderately strong battery and is made to pass once more over the wire the magnetic writing is obliterated under the influence of the constant magnetising force, which is great compared with the writing magnetic forces."

And in describing his "Phonographic Newspaper" reading machine, *Fig.* 7, Poulsen said:

"A long steel ribbon is stretched between two rolls which can rotate at considerable speed. The ribbon passes a series of electromagnets at a speed regulated according to circumstances. The electromagnet, *E*, inscribes words, music, etc.; the other electromagnets —'the reading magnets'—reproduce the communications in the telephone of each hearer; and, finally, the obliterating magnet, *V*, equalises the magnetic variations of the ribbon. As using does not weaken the writing, we are able to intercalate any number of reading magnets."

Wide use of the magnetic recorder invention was delayed until better wire, such as 18-8 stainless steel (which becomes highly magnetic when cold-worked) and more practical mechanical reeling arrangements were worked out by others in Europe, the United States and Japan. It was apparent from the beginning that here was an inherently simple way of making everlasting recordings of high fidelity and low bulk and cost, as much as one hour playing time now being recorded for each cubic inch of wire.

Oberlin Smith not only anticipated the Poulsen Wire Recorder of 1900 but also the more recently developed paper-backed magnetic tape, first used by Pfleumer in the magnetophone in Hitler's Germany. Both Edison and Smith suggested many lines of the phonographic invention that have not been followed, and which, in view of the tremendous size of the sound industry and the new electronic facilities now available, might well repay re-examination.

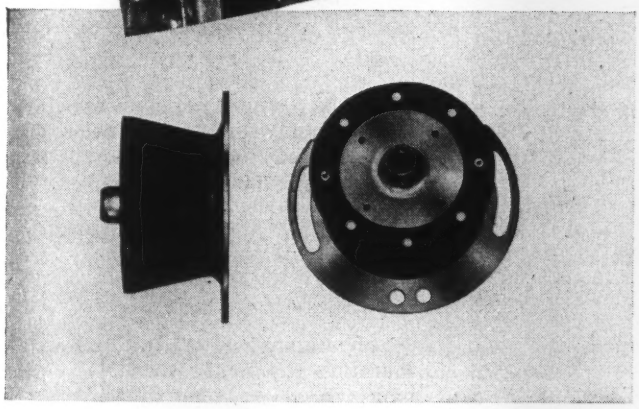
Extrusions Solve Winding Problem

High conductivity, soft annealed copper bar, 23,000 ft of it, was required for use as magnet winding for the "Cosmotron" atom smasher at the Brookhaven National Laboratory, Associated Universities Inc. Two sizes of bars were needed, 2.182 x 0.75-inches and varying in length to 52½ ft, and 1.875 x 0.75-inches in lengths to almost 48 ft. The problem consisted of getting a cooling-water hole, having an area of 0.0941-sq inch, through the bars within a ½-inch circle at the section's center. The Buffalo Branch of the American Brass Co. successfully produced the required 150,000 lb of these special sections on a 2500-ton extrusion press.

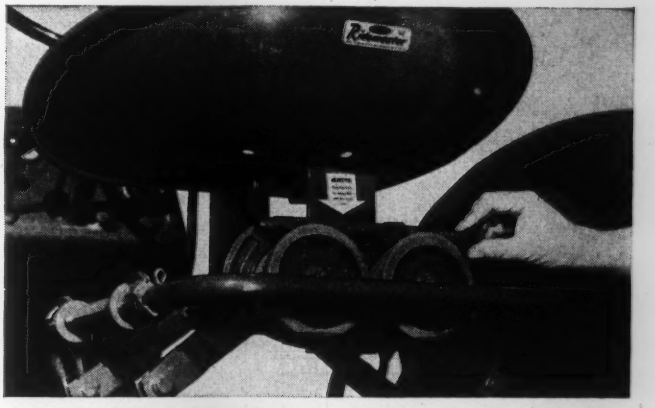
NEW RUBBER SPRING PROVIDES UNIQUE BENEFITS

Used on new tractor seat, it eliminates slow-down from rider fatigue . . . increases production

GOOD-BYE TO "ROUGH RIDING." As the tractor rides over rough ground, the seat's rubber torsional springs twist and turn with each jolt, cushioning shock and leveling out the sharp rebound. Result: less fatigue for the farmer . . . greater working efficiency . . . more daily output from the tractor in the same amount of time!



THE PRINCIPLE OF THE THING. These stabilizing springs operate on a torsional principle . . . consist of a thick layer of carefully compounded rubber, sandwiched between metal plates. Springs were especially developed by U. S. Rubber engineers for the manufacturers of this revolutionary new seat.



SMOOTH SEAT FOR EVERYONE. A twist of this tie bolt and the seat spring can be adjusted to provide the utmost comfort for the slimmest farm girl or the huskiest male. Another good example of the results obtained by manufacturers who bring their problems to U. S. Rubber's engineering staff!

Hundreds of different products, from baby carriages to railway cars, can be improved by the use of the new U. S. Rubber Torsional Spring. It may well be that your own product, whether on the market or in the design stage, can obtain with this spring the competitive advantage it requires for real success. The tractor seat above is only one instance of how this versatile development aided a manufacturer. Ask "U. S." engineers to investigate its possibilities for your product. Write

A PRODUCT OF

U.S.RUBBER

SERVING THROUGH SCIENCE

UNITED STATES RUBBER COMPANY

MECHANICAL GOODS DIVISION • ROCKEFELLER CENTER, NEW YORK 20, N. Y.

SPECIFY PARKER
TRIPLE-LOK
THE J.I.C.
TUBE FITTING

- 37° flare angle
- Dryseal pipe threads
- has *actually* passed rigid AN-F-47 performance test as required by J. I. C. standards. *Certified laboratory test report* available.

MADE BY THE WORLD'S LARGEST
MANUFACTURER OF LEAKPROOF TUBE FITTINGS.

- precision-made to be *leakproof* under even the severest operating conditions.
- holds *leakproof* beyond tube bursting pressures.
- 3-piece design—remains *leakproof* after repeated reassembly.
- accommodates full range of standard tube wall thicknesses.
- made in steel, stainless steel, also aluminum alloy.
- shape bodies machined from forgings, other parts from bar stock.
- stocked by distributors across the nation.
- write for Parker "Industry Standard" Tube Fittings Catalog 203.

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5827 W. CENTURY BLVD., LOS ANGELES 43, CAL.

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The PARKER Appliance Company
Cleveland 12, Ohio

Please send my free copy of Tube Fittings Catalog 203

Name_____

Company_____

Address_____

City_____ State_____

SALES AND SERVICE PERSONNEL

RECENTLY APPOINTED assistant sales manager for Ward Leonard Electric Co., Mount Vernon, N. Y., **W. C. Schofield** will assist **Frank E. Beede,** company sales manager. Upon graduation from the University of Maryland in 1930, Mr. Schofield completed the student course at Westinghouse Electric Corp., East Pittsburgh, Pa., and then served as Westinghouse sales engineer for the Washington, D. C., area for six years. In 1938 he established the Federal Engineering Co. in Washington, D. C., and was co-owner of this company until 1948. Since that time he has been directing sales in the industrial control division of Ward Leonard Electric Co.

♦

Formerly manager of the Detroit office of the Lord Mfg. Co., **Charles L. Freel** has been made administrative assistant to general sales manager **George E. Tubb.** His headquarters will be in Erie, Pa., at the home office of the company. **Everett C. Vallin** has replaced Mr. Freel in Detroit as representative in the Michigan and northwestern Ohio territory.

♦

John C. McLeod has been appointed manager of Rusco belting sales, Chicago branch, for the Russell Mfg. Co. He will handle the complete line of Rusco high-speed endless belts and transmission and conveyor belting for the Midwest territory.

♦

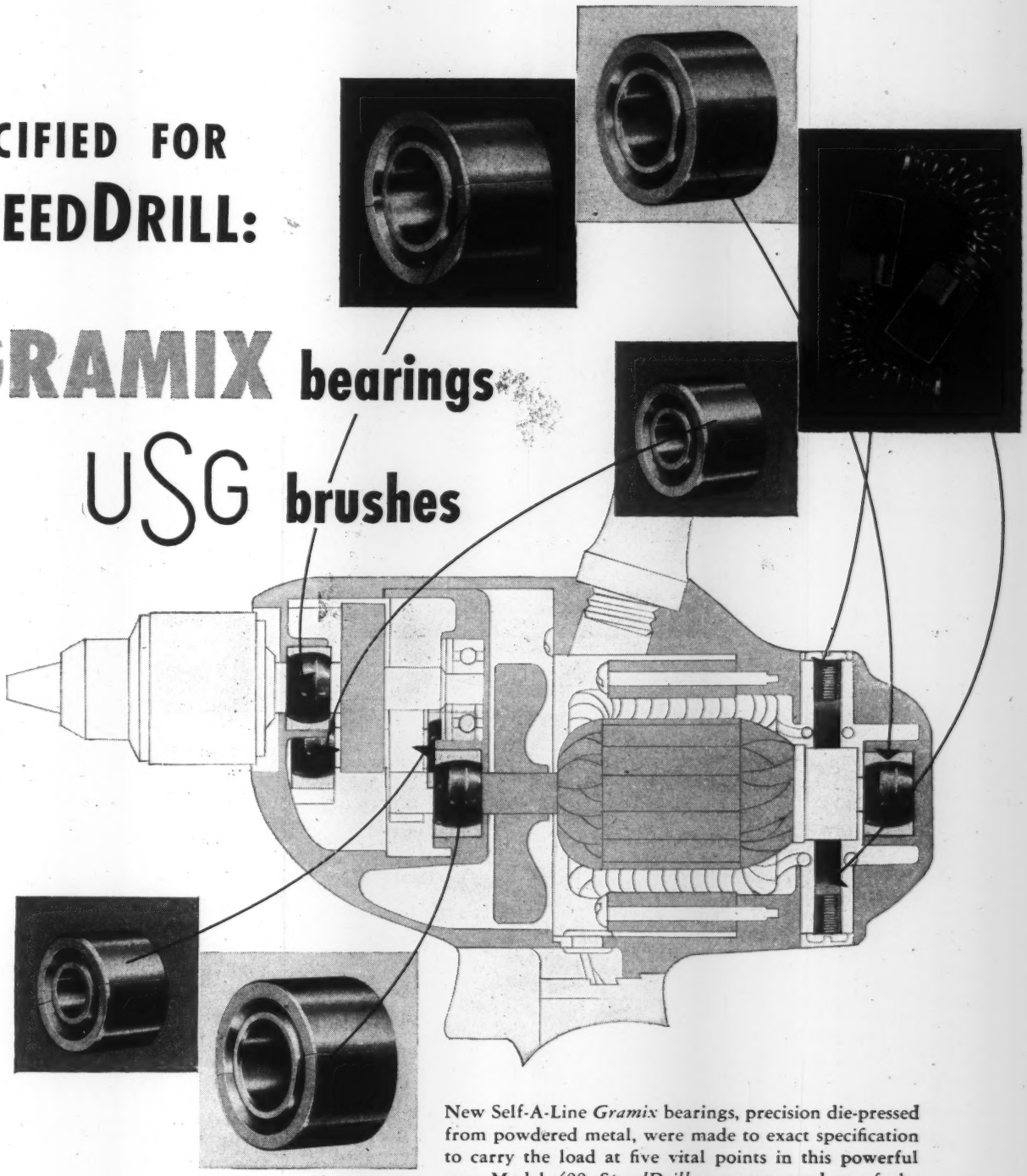
Sales manager of the company's materials handling division for the past ten years, **John C. Mevius** has been appointed to the post of sales manager of the hydraulic division of American Engineering Co., Philadelphia. In addition to his new duties, he will continue to be responsible for materials handling sales and will be assisted in hydraulic sales by **G. A. Fleck.**

♦

Several organizational changes in the mechanical goods sales division of Goodyear Tire and Rubber Co. were announced recently. **R. C. Stein,** who joined Goodyear in 1943, has been promoted to manager of molded goods sales at the St. Marys, O., plant. **R. E. Chapman,** who entered the mechanical goods organization in Chicago in 1937, was appointed manager

SPECIFIED FOR SPEEDDRILL:

GRAMIX bearings
USG brushes



New Self-A-Line *Gramix* bearings, precision die-pressed from powdered metal, were made to exact specification to carry the load at five vital points in this powerful new Model 400 *SpeedDrill* a recent product of the SpeedWay Manufacturing Company of Cicero, Illinois. Unique design of the *Gramix* bearings enables them to maintain steady alignment of the drill shafts and to absorb the thrust load under hardest operating conditions. The self-lubricating features of Gramix eliminates need for regular oiling, and the great durability of these bearings makes for long life in any application. Teamed with *Gramix* in the *SpeedDrill* are two special USG carbon brushes which will furnish positive balance of brush pressure on the commutator at all drill speeds.

Send your prints to us — our engineers will be glad to make recommendations without obligation. Write for the new *Gramix* and USG Brush catalogs.

GRAMIX

76

THE UNITED STATES GRAPHITE COMPANY, SAGINAW, MICH. DIVISION OF THE WICKES CORPORATION

GREER ACCUMULATOR
Eliminates 9 hrs. Machining Time
Saves $90 in Cost and 20% in Floor Space of Brosius Manipulator

Photo Courtesy of:
Edgar E. Brosius Co., Inc.
Pittsburgh 15, Pa.

The huge Manipulator shown here weighs 31,000 lbs. and handles steel billets or blanks up to 12,000 lbs. Yet it moves about with amazing agility. It draws or charges, tilts up or down, raises or lowers, rotates and moves from side to side with the ease of a light-weight and will bring to your forge shop added speed, safety and economy.

Mr. S. T. Morgan, Chief Engineer of the Brosius Company reports that by incorporating a Greer Accumulator in its design a saving of 9 hrs. of machining time, 1 hr. of assembly time, and a 20% of floor space was easily obtained. And in addition, the cost of the manipulator was cut $90.

This is but one of numerous applications where Greer Accumulators are reducing cost, improving efficiency and enhancing safety and economy of systems and machines using hydraulics for power transmission.

Our bulletin 300 describes in detail the construction and uses of Greer Accumulators. Write for your copy on your company letterhead today.

SALES REPRESENTATIVES IN ALL PRINCIPAL CITIES

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GREER
HYDRAULICS INC.
454 EIGHTEENTH ST BROOKLYN 15, N.Y.

YOUR SYMBOL OF SERVICE

of hose sales, Akron, succeeding **R. W. Sanborn,** who is now representing the division in the Akron territory. Mr. Sanborn has been with the company for thirty-four years. The company has also announced the appointment of **C. A. Hulsemann** as manager of the industrial brake sales department, aviation products division. This new department was established to handle increasing demand for the revolutionary industrial disk brake, adapted in 1948 from the company's airplane-type, single-disk brake for industrial use.

♦

The appointment of **Robert E. Ward** and **Ralph D. Ferguson** as manager and sales manager, respectively, of the foundry at the Eclipse-Pioneer division of Bendix Aviation Corp. was announced recently. Mr. Ward, previously director of metallurgy for Eclipse-Pioneer, has been with the division since 1938. Mr. Ferguson formerly was assistant manager of the foundry and has been with Eclipse-Pioneer since 1932.

♦

Michiana Products Corp. of Michigan City, Ind., maker of alloy castings and high temperature fans, has announced several additions to its sales force. **H. B. Scott,** previously with Calorizing Corp. as metallurgical engineer, is in charge of the southern Indiana area with headquarters at 310 Test Bldg., Monument Circle in Indianapolis; the Philadelphia area has been assigned to **L. S. Johnson,** 637 West Sedgwich St.; and **A. J. Popovich,** formerly metallurgical engineer with Allis-Chalmers Mfg. Co., is now in charge of the northern Indiana territory with headquarters at the plant.

♦

According to a recent announcement, **Mert M. York,** who joined Allis-Chalmers Mfg. Co in 1939, has been named manager of the company's Boston district office.

♦

Park Q. Wray, manager of the replacement sales division of National Motor Bearing Co. Inc. since 1938, has been advanced to the company's key sales post as director of sales for the company and its two subsidiaries, Arrowhead Rubber Co. and National Seal Co. **E. F. Riesing,** veteran sales executive of the automotive parts division of Firestone Tire and Rubber Co., has been appointed sales manager of the industrial sales division, and **Park Wray Jr.,** for some time in charge of the northern California sales territory of the replacement division, has been promoted to

"Commercial" Ball-Bearing Data...#8 of a Series

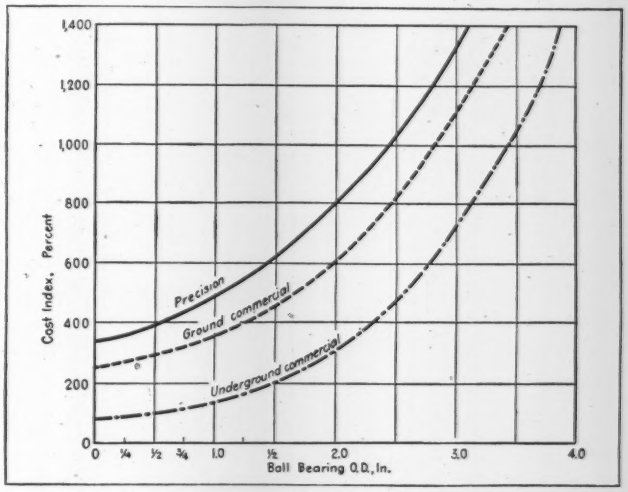
WE WILL BE GLAD TO SEND YOU COPIES OF THE PREVIOUS ADVERTISEMENTS IN THIS SERIES.

Schatz "Commercial" Ball Bearings can reduce anti-friction bearing costs 35 to 85%

Expensive precision bearings with their extremely close tolerances are often unnecessary in certain applications.

Where the operating conditions of these applications are within the load and speed limits of Schatz "Commercial" Ball Bearings, you can obtain the required anti-friction bearing performance—at ⅔ to ⅙ the cost of precision bearings.

The savings possible in such instances are illustrated in the graph at the right.



Relative costs of precision and "Commercial" Ball Bearings. In very small sizes, the cost of precision bearings *increases* as the bore decreases, due to the difficulty in maintaining close tolerances. The cost of "Commercials" *decreases* in almost direct proportion to the size.

Schatz "Commercial" Ball Bearings are classified as follows:

UNGROUND—FOR LIGHT LOAD, LOW-SPEED OPERATION—the lowest-cost ball bearing available.

GROUND—2.17 TIMES THE RADIAL LOAD-CARRYING CAPACITY OF UNGROUND TYPE. Inner and outer races are ground to superfine finish for smoother, faster operation and heavier loads.

SPECIAL BEARINGS—ENGINEERED TO SPECIFIC NEEDS—often at no greater cost than standard types. Various modifications—in materials, inner ring, outer ring and ball arrangement—make it possible to use a special bearing where no standard type can fill the requirements. Costly machining and assembly time required for bushings or plain sleeve bearings is eliminated.

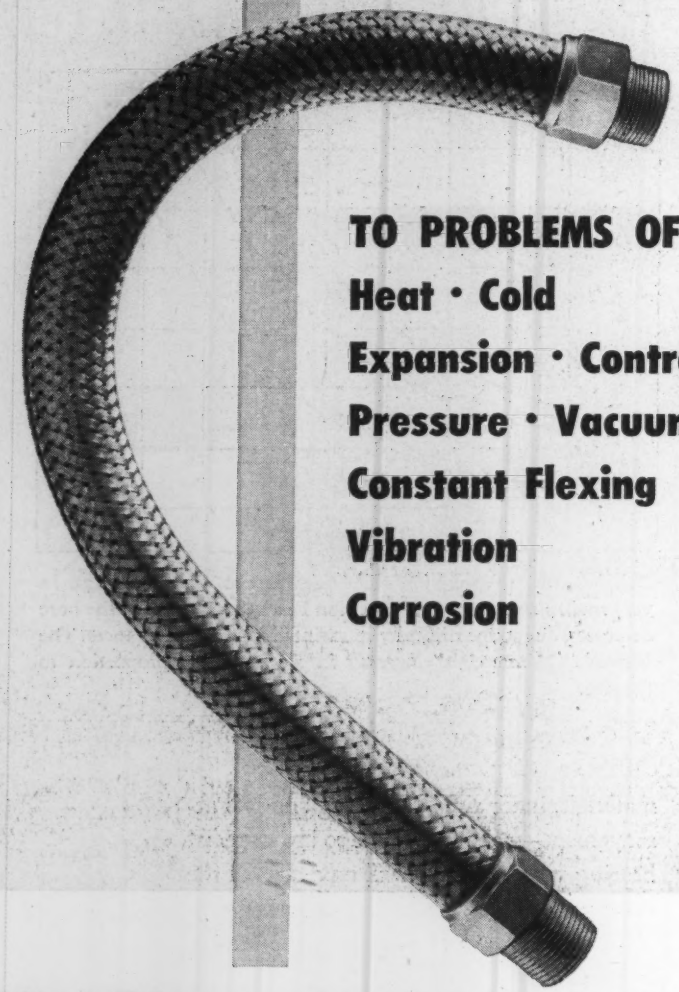
All Schatz "Commercial" Ball Bearings, single or double row, can be equipped with felt or labyrinth seals, flanged or plain outside diameters, hardened or unhardened treads, set screws, Alemite fittings, extended inner ring and hexagonal bore.

Write for your copy of "Construction and Characteristics of Low-Cost Ball Bearings." It tells how to select "Commercial" Ball Bearings for various speed and load requirements, gives the classifications of standard types for radial and thrust loads, and examples of the modifications that can be made for special design, assembly or operating conditions. Ask for Catalog No. 11, too. It gives complete data for the whole "Commercial" line.

SCHATZ
"Commercial"
BALL BEARINGS

The Schatz Manufacturing Company
6760 FAIRVIEW AVENUE, POUGHKEEPSIE, N. Y.

Economical Answer



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Constant Flexing
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- ★ Stays flexible and stays tight under severe conditions

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TITEFLEX FILLS YOUR TUBING REQUIREMENTS TO A . . . T

sales manager of the division. Park Q. Wray and Park Wray Jr. will have their offices at the company's plant in Redwood City, but Mr. Reising will headquarter in the Detroit office in order to be close to the company's many industrial customers in the Chicago-Detroit-Cleveland area.

♦

With offices at 7-233 General Motors Bldg., Detroit, **R. R. Rolph** has been made Detroit sales representative of the Warner Gear division of Borg-Warner Corp. For many years Mr. Rolph was vice president in charge of automotive sales for the Mechanics Universal Joint division of Borg-Warner, with headquarters in Detroit, and since 1947 he served as sales manager of the Monroe Auto Equipment Co.

♦

Eutectic Welding Alloys Corp., New York City, manufacturers of Eutectic Low Temperature Welding Alloys and fluxes, has made a series of field force promotions headed by the appointment of **Don Rasmussen** as western sales supervisor. Other advances announced were those of **Herman Greif** and **Daniel Wetmore** to regional managers; **Hugh Hurley** and **Roy Waldrop** to assistant regional managers; and **Don Garrett, Lee Wade** and **Alex Frederick** to district managers. These men will now supervise the activities of the group of Eutectic district engineers in their various areas.

♦

George Carleton Jr. has recently been made president of the Nice Ball Bearing Co., Philadelphia, succeeding **W. H. Rossmassler,** who has retired. Mr. Carleton was first employed in the company's sales department in 1929 and since then has been successively advanced to the positions of sales manager, vice president, executive vice president and president.

♦

Kennametal Inc., Latrobe, Pa., manufacturers of cemented carbide tools, has announced the appointment of **Thomas J. Kniff Jr.** as representative in the New York office, 11 West 42nd St., and **Fred Moore** as representative in the central district working out of the Detroit office at 5531 Woodward Ave.

♦

In a further step to extend its field organization, the American Pulley Co. of Philadelphia has appointed **Gerard P. Weishaar** as Eastern regional manager. He will have direct charge of district managers and representatives in the New England, Rochester,

NOW 8 BASIC CHRYSLER ENGINES

FROM WHICH TO CHOOSE

In scores of industrial applications, Chrysler Industrial Engines are out-performing all other similarly rated engines. With less down time! With lower lifetime cost per hour. That comes from famed Chrysler engineering and high precision manufacturing. Yet thanks to Chrysler mass production economies, you can buy Chrysler power for *less money!*

If you use internal combustion power requiring 200 to 400 cubic inches piston displacement, get *more for less!* Phone your Chrysler Industrial Engine dealer or write us. ***Industrial Engine Division, Chrysler Corporation, Detroit 31, Michigan.***

CHRYSLER

Industrial Engines and Power Units

HORSEPOWER WITH A PEDIGREE

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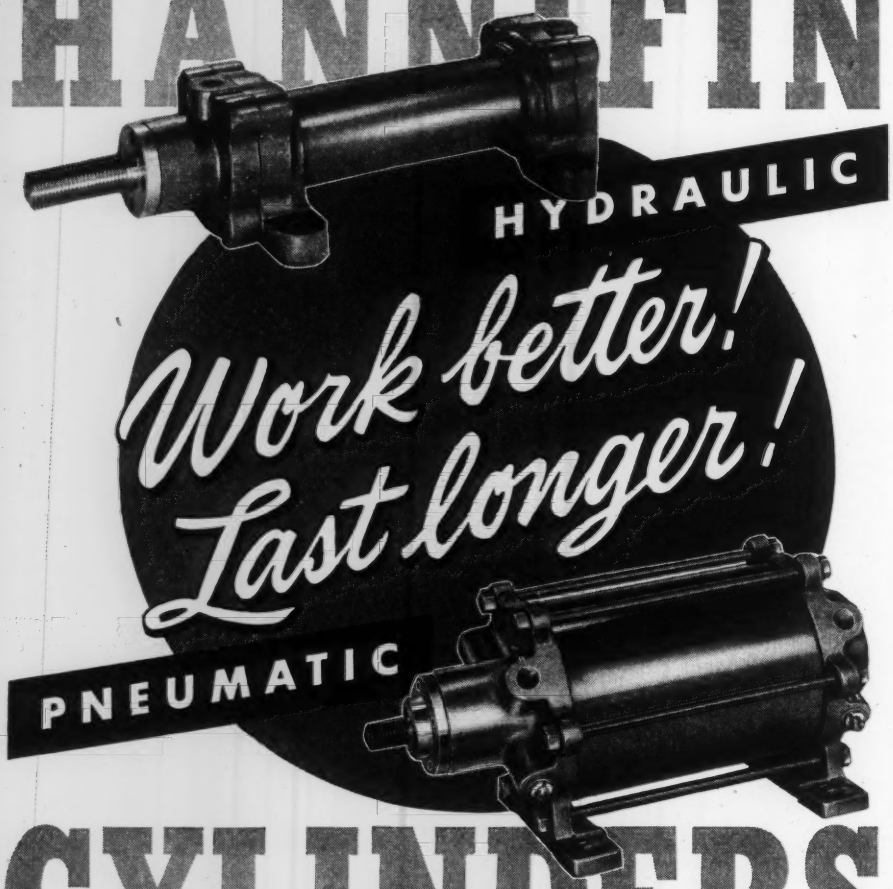

CYLINDERS

A BIG ADVANTAGE in using Hannifin Cylinders is the way they *last!* Such performance is of prime importance in installations where shut-downs for repairs would cost far more than the repairs themselves. This is one reason why Hannifin Cylinders have long been the choice of builders of heavy duty production equipment and precision machine tools.

STANDARDIZED FOR LOWER COST... QUICKER DELIVERY!—Hannifin Air Cylinders are offered in 10 standard sizes, from 1" to 12", and 6 standard mounting styles. Hannifin Hydraulic Cylinders are offered in 12 standard sizes, from 1" to 8", and 12 standard mounting styles. Any length stroke; single or double end rods; adjustable cushions for head end, rod end, or both. Many combination mountings available. Special cylinders built to meet any requirements.

COMPLETE LINE!—You can save time and money right from the start by checking with Hannifin on all of your hydraulic and pneumatic cylinder requirements. The Hannifin line is complete!

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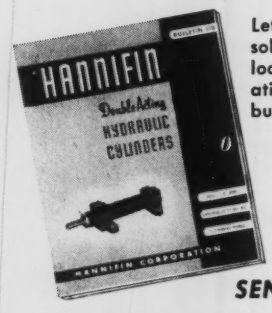
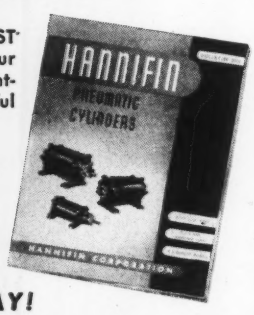


Let Hannifin engineers help you get the BEST solution for your cylinder problems. See your local factory-trained Hannifin field representative today. Ask for copies of the helpful bulletins illustrated here:

- **Bulletin 110 (left) "HANNIFIN HYDRAULIC CYLINDERS."**
- **Bulletin 210 (right) "HANNIFIN PNEUMATIC CYLINDERS."**



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AIR CYLINDERS • HYDRAULIC CYLINDERS • HYDRAULIC PRESSES
PNEUMATIC PRESSES • HYDRAULIC RIVETERS • AIR CONTROL VALVES

New York, Philadelphia, Charlotte, and Atlanta territories. In this new capacity, Mr. Weishaar's operations will parallel those of **Randolph Jackson,** who is Midwest regional manager, with headquarters at 332 South Michigan Ave., Chicago 4, Ill. At the same time, the company announced the appointment of **John T. Barron** as manager of its power transmission division. His activities will be centered on the product and market development of American flat belt, V-belt, and speed-reduction drives and other types of mechanical power transmission.

♦

Synthane Corp. of Oaks, Pa., laminator and fabricator of thermosetting plastics for industry, has appointed **Gene Rankin** as sales representative in the Kansas City area to succeed the late **Ernest Bernard.** Mr. Rankin, who had been Mr. Bernard's assistant since 1946, will direct sales in Nebraska, Kansas, Colorado, western Missouri and southwestern Iowa.

♦

G. W. Fine has been named assistant sales manager of U. S. Stoneware Co.'s plastic coatings division. Mr. Fine was associated with Sherwin Williams Co. for seven years, first as a laboratory technician and later as a member of the sales staff.

♦

Previously service manager, **Millard S. May** has been named sales manager for the Speer Carbon Co., St. Marys, Pa. At the same time several appointments were announced in the International Graphite and Electrode Corp., a Speer subsidiary. **John W. Lohnes,** formerly manager of sales for International Graphite, has been made vice president in charge of carbon and graphite sales for both the parent company and International Graphite. Robert E. Rice, formerly assistant sales manager of Speer Carbon, is now sales manager of International Graphite. **William E. Harvey** who was plant manager for International Graphite at Niagara Falls, N. Y., has been appointed plant manager of all carbon and graphite plants of the Speer Carbon Co. and its subsidiaries.

♦

The Federal Machine and Welder Co., Warren, O., manufacturers of Federal resistance welders and Warco mechanical presses, has announced that its New York office is now under the direction of **Carl de Bourbon** of Carbet Inc., 111 Broadway New York, N. Y. Federal was formerly represented by the A. A. Probeck Co., in New York.

IT'S A GROWING MOVEMENT . . .

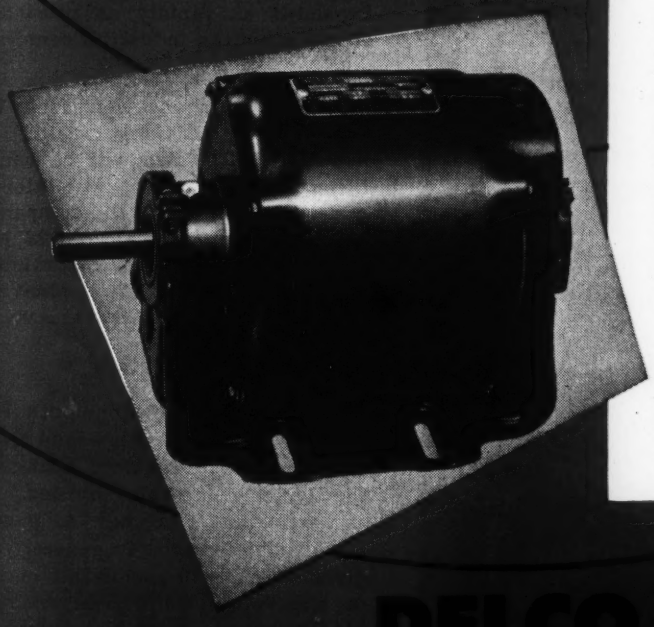
DELCO PREFERRED

FOR QUALITY MOTORS

FLEXIBLE SERVICE

THERE IS A DELCO MOTOR FOR:

Refrigerators	Compressors
Washing machines	Home work shops
Dryers	Fans (ventilating)
Furnace blowers	Dishwashers
Oil burners	Ironers
Stokers	Office machines
Water pumps	Feed grinders
Gasoline pumps	Separators



Appliances need good motors to give dependable service. Years of engineering and manufacturing experience have put top quality into Delco appliance motors.

But today, appliance manufacturers need even more than that. Fast changing markets often mean changes in motor specifications or delivery dates. Because Delco is organized to meet these requirements, manufacturers get what they want—when they want it—changes included.

That's why more and more spec sheets carry the note, "DELCO PREFERRED." Compact Delco Motors, designed to meet the specific torque and service requirements of the application, are built in sizes from 1/8 H.P. up.

✓ ✓ ✓

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Division of General Motors Corporation, Dayton, Ohio

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OVER ONE HUNDRED YEARS OF CONTINUOUS SERVICE. ROUNDS, SQUARES, FLATS, HEXAGONS, OCTAGONS

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Get your STANDARD ALLOY STEEL from the same reliable source as HY-TEN ALLOY STEEL

Although perhaps best known for our *special* HY-TEN Alloy Steels, Wheelock, Lovejoy carries a full line of *standard* steels *in stock* for fast, dependable service from our warehouses. These standard grades include: C-1117, A4615, E4617, A4620, A4140, A4142, A4145, A4150, A4340, etc.

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WL *steels are metallurgically constant. This guarantees uniformity of chemistry, grain size, hardenability — thus eliminating costly changes in heat treating specifications.*

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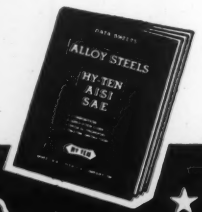

SALES NOTES

NEWLY APPOINTED distributor in southern Wisconsin for the complete line of industrial rubber products manufactured by the **United States Rubber Co.** is the **F. D. Haker Co.,** 713 South Tenth St., Milwaukee, Wis. Acquisition of the U. S. Rubber line will enable the Haker Co. to offer southern Wisconsin a complete sales and technical service for such products as power transmission belting, elevator and conveyor belting, all types of industrial hose and packing, grinding wheels, and many molded rubber specialties.

♦

Recent appointment of distributors stocking **Blackmer Pump Co.'s** entire product line and service parts was announced recently. Included are **Blott-Robb Co., Transmission Machinery Co.** and **Amco Corp.,** all in Chicago, **H. O. Link Co.** in Baltimore, **Tri-State Equipment Co.** in Kansas City, and **Avels Sales and Engineering Co.** in Indianapolis. This newly instituted stocking service includes engineering assistance which will be expanded as rapidly as possible to all metropolitan industrial centers.

♦

A recent announcement from **Wolverine Tube Division** stated that henceforth all sales will be handled through the new divisional sales and advertising offices of the company at 1850 Guardian Bldg., Detroit 26, Mich. Sales to wholesalers will continue to be handled by the Detroit plant sales department at 1411 Central Ave., Detroit 9, Mich.

♦

The name of the Monmouth Products division of the **Cleveland Graphite Bronze Co.** has been changed to the Replacement Sales division. This division thus becomes a part of Cleveland Graphite's general sales division and will handle all markets in the replacement field, including NAPA sales, export sales and special markets. **Raymond Z. Oswald,** vice president of replacement sales, heads the replacement organization. **J. E. Bradley has been named sales manager, distributor sales, with R. H. Wickersham** assistant sales manager of distributor sales. **A. M. Currier Jr.,** is sales manager, special mar-

IMPORTANT SAVINGS gained by Users of PERMITE ALUMINUM CASTINGS

USERS of Permite Aluminum Castings save on machining costs because all Permite Castings are made to close tolerances that reduce machining time.

They save on production time because Permite Castings are always uniform in quality. Permite modern facilities and strict quality control result in castings that are right in dimensions, in strength, in ductility, in corrosion resistance — right in every detail essential to the customer's exact requirements. Whether your aluminum castings are made by the permanent mold, semi-permanent mold, sand cast, or die cast process, you can count on Permite Aluminum Castings for the qualities you expect.

Ask for Permite recommendations and quotations on your castings requirements.

Ask for 80-page Castings Manual . . .

Authoritative, illustrated manual on the design, production and uses of aluminum castings. Includes casting alloys tables. Sent free on request to interested executives.

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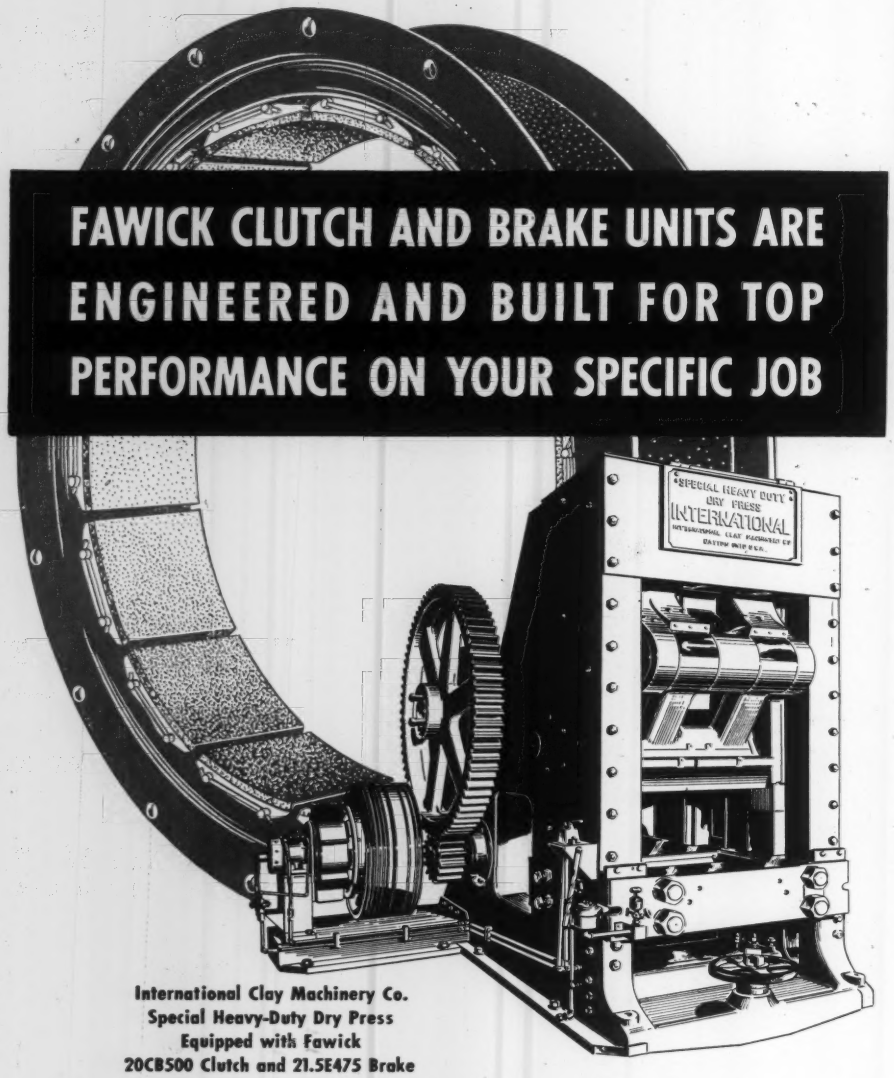
ALUMINUM INDUSTRIES, INC.

CINCINNATI 25, OHIO

ALUMINUM PERMANENT MOLD, SAND and DIE CASTINGS...HARDENED, GROUND and FORGED STEEL PARTS

FAWICK CLUTCH AND BRAKE UNITS ARE ENGINEERED AND BUILT FOR TOP PERFORMANCE ON YOUR SPECIFIC JOB



International Clay Machinery Co.
Special Heavy-Duty Dry Press
Equipped with Fawick
20CB500 Clutch and 21.5E475 Brake

When you choose Fawick clutches and brakes for your machines, you know that every application is designed and built to meet your specific operating requirements.

With Fawick on the job, you get the clutch and brake that *does the work you want done.*

The simple rugged design and construction of Fawick units, plus the elimination of lubrication, mechanical linkage and adjustment, provide long maintenance-free life in heavy-duty service.

These are some of the reasons why you can step up production and lower maintenance costs with Fawick.

For specific information on all advantages of Fawick Industrial Clutch and Brake Units, call or write the Main Office, Cleveland, O., for Bulletin 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK CO., INC. Airflex
9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = AXIAL CONTACT + 360° RADIAL CONTACT + SHOCK ABSORPTION + CONTROLLED TORQUE = PEAK EFFICIENCY

kets, and W. M. Williams is now assistant to the vice president of replacement sales. All were formerly associated with Monmouth Products Co., which was acquired last year by the Cleveland Graphite Bronze Co. The Replacement Sales division will merchandise its line of engine bearings, bushings, clutch parts and front wheel parts to the NAPA and export trade under the established Monmouth brand name, as in the past, and headquarters of the division will continue in the former Monmouth plant at 6545 Carnegie Ave., Cleveland, which is now the Carnegie plant of the Cleveland Graphite Bronze Co.

♦

Opening of a direct sales office in the ASTE Bldg., 10700 Puritan Ave., Detroit, Mich., was announced recently by the **Taft-Peirce Manufacturing Co.** Raymond F. Jalbert, district manager of the Detroit territory, has been assigned to the office, and he will be assisted by Richard Read, also of the Taft-Peirce sales force.

♦

The Cleveland office of the **Robins Conveyors Division, Hewitt-Robins Inc.,** has moved from 215 Rockefeller Bldg. to 8905 Lake Ave.

♦

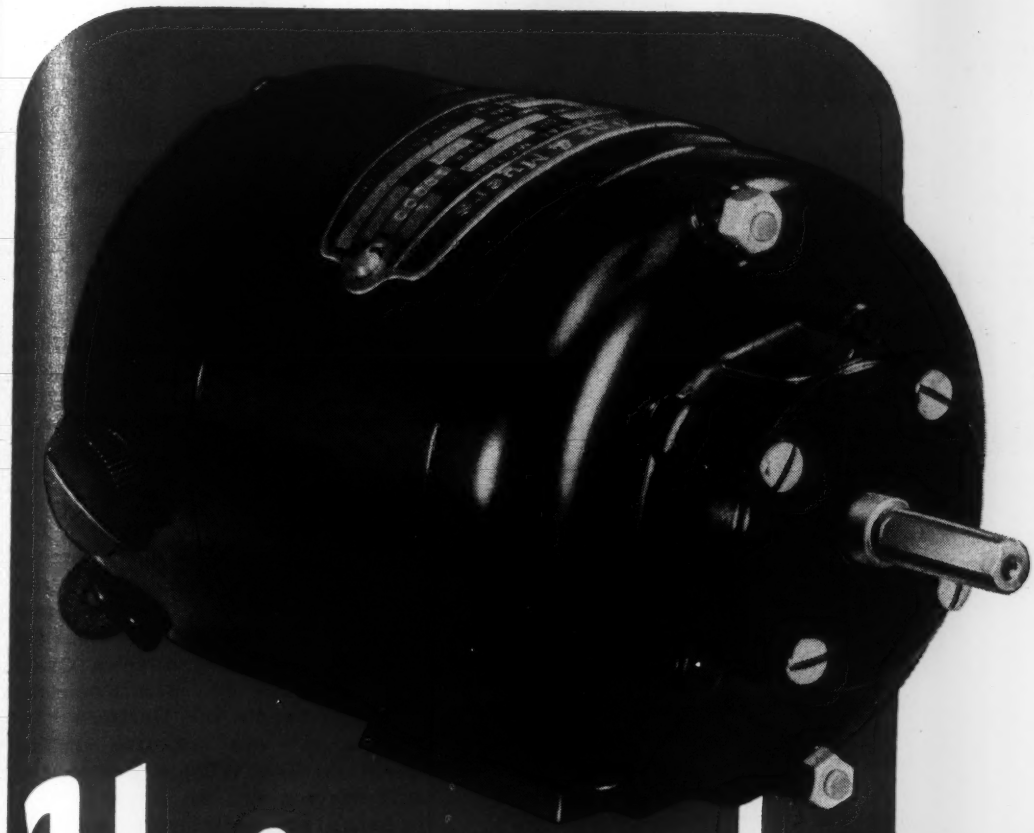
With sales and engineering offices at 1125 Arbor Ave., Dayton, O., and 3091 Mayfield Rd., Cleveland, in addition to its main office at 451 West Market St., Akron, **Kepler Engineering Co.** has been appointed exclusive representative for **Hydro-Line Manufacturing Co.'s** air and hydraulic cylinders in the state of Ohio and northern Kentucky.

♦

Removal of its office and plant to Hawley Rd., Mundelein, Ill., has been announced by the **Hubbell Corp.,** manufacturers of valves, controls and regulators.

♦

Announced recently was the reorganization of **American Smelting and Refining Co.'s** sales and distribution plans for continuous-cast products of brass and bronze. These products were formerly distributed through **Ampco Metal Inc.,** Milwaukee, as general sales agent. John L. Kimberley has been named sales manager, Continuous-Cast Products, and will direct the new program from the Perth Amboy plant. Ralph W. Bailey and John V. Hackett have been appointed assistant sales managers. Donald S. Goebel, who handled the Asarco continuous-cast alloys



Universals by R&M

● New R & M Universal Motors are something *you* should know about. To inherent *"series"* characteristics—high starting torque; high operating speed; adaptability to speed control—Robbins & Myers has added *many improvements* in performance and appearance.

BETTER, INSIDE AND OUT . . . From their dynamically balanced armatures to their rigid, welded-steel shells, *nothing* has been slighted. Commutation, brushes, bearings, insulation, and ventilation all represent progressive foresight, *backed* by long experience.

YOUR CHOICE . . In the range of 1/100 to ⅓ horsepower (covered by four frame sizes), there's the right motor for *every* "universal" service—sleeve or ball bearing; fixed rotation or reversible; open or totally enclosed; standard or special rating—and a *welcome variety* of housing types and end-heads.

GET THE FACTS ON THESE BETTER MOTORS . . . For quality motors with a *modern plus,* call in Robbins & Myers. Literature on request. And, if you would like to have R & M sit-in on a discussion of *powering improvements,* that's yours for the asking, too. R & M motors are *good* motors. Address Dept. E-80.

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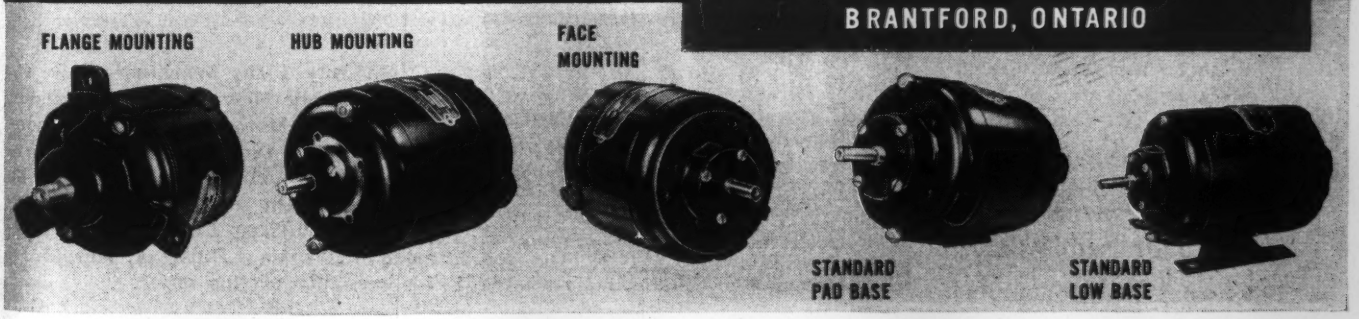
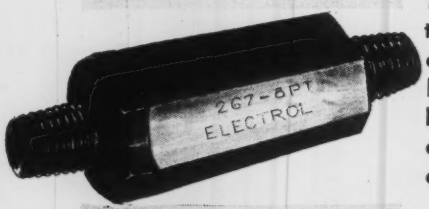


THE 523 RELIEF VALVE is compact in design, light in weight, neat in appearance, and can be easily and quickly installed.

No matter whether they're employed to meet the exacting requirements of a maker of X-Ray equipment, or measure-up to the heavy-duty demands of a machine tool builder — Electrol Relief and Check Valves provide precise pressure and flow control . . . *and work quietly together.*

The Electrol Relief Valve is of poppet-type construction, which assures fast, accurate, squeal-free pressure control. It has a wide range of adjustment, and can be set to the desired position, easily and quickly, by turning a fine, screw-thread adjusting cap.



THE 267 CHECK VALVE is for use in air or oil at operating pressures up to 1,500 p.s.i. It is of aluminum alloy construction, and is designed especially for small space installations.

Electrol Check Valves are designed to eliminate any possibility of excessive pressure drop, and are equipped with a patented O-ring check seat, which provides a quick acting, positive check of reverse flow from 0 up to the maximum operating pressure.

Let us supply you with further details, or —if you wish—have one of our engineers call at your convenience.

THE 463 CHECK VALVE is for use in air, oil or water at operating pressures up to 5,000 p.s.i. It is of brass and bronze construction, and is highly adaptable to heavy-duty service.

Better Designed Products Use Electrol Hydraulics

Electrol
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89 GRAND STREET
KINGSTON, NEW YORK
CYLINDERS • SELECTOR VALVES • FOLLOW-UP VALVES
CHECK VALVES • RELIEF VALVES • HAND PUMPS
POWERPAKS • LANDING GEAR OLEOS • SOLENOID
VALVES • ON-OFF VALVES • SERVO CYLINDERS • TRANSFER
VALVES • CUT-OUT VALVES • SPEED CONTROL VALVES
FOR BETTER HYDRAULIC DEVICES

when they were distributed by Ampco Metal Inc., has joined American Smelting and Refining Co. and will make his headquarters at the Whiting, Ind., plant. American Smelting representatives will handle the sales for 37 states in the eastern, midwestern, south and southwestern areas, with the western area being handled by **Kingwell Bros. Ltd.** of San Francisco.

♦

The **Hartford Special Machinery Co.**, Hartford, Conn., has announced the appointment of the following representatives: **Dayton & Bakewell,** 1939 Santa Fe Ave., Los Angeles, Calif., to handle swagers, die polishers and "Super-Spacers"; **Hoffman & Heartt,** 3005 South Grand Ave., Los Angeles 7, Calif., to handle automatic drilling and tapping machines and the Hartford thread roller; **Thomas Wallace,** 711 Architects Bldg., 806 West Fifth St., Los Angeles, Calif., and 320 Tenth St., San Francisco, Calif., to handle the Hartford "Super-Spacer" and sensitive drilling machines; and **William Scheer Co.,** 6376 Clayton Rd., St. Louis 17, Mo., to handle automatic drilling and tapping machines, thread roller, die polishers and sensitive drilling machines.

♦

Recently appointed exclusive representative for the state of California for the **United States Gasket Co.** of Camden, N. J., **W. S. Shamban & Co.,** process equipment sales engineers in Los Angeles, offers a complete line of gaskets and packing including Chemiseal Teflon products.

♦

Plans for the construction of a new office and warehouse have just been announced by the **Chicago Steel Service Co.,** middle west distributor of stainless and carbon steels, with headquarters in Chicago. The new building will be located at 45th St. and Kildare Ave. in the Crawford development of Chicago's central manufacturing district. Plans call for the construction of a one-story, steel-frame structure with 120,000 square feet of floor space, to be completed in late fall.

♦

On July 1 the Washington, D. C., office of the **New Departure Division** of **General Motors Corp.** was discontinued. The territory previously covered by the office will henceforth be served by the Philadelphia zone office at 850 East Luzerne St., Philadelphia 24, Pa. Harry A. Herold Jr. is manager of this office.

Whether you want one worm-gear speed reducer or a thousand, a standard drive or a unit designed to meet special problems, Cleveland can serve you better.

Prompt service on dependable worm-gearing

For dependable worm-gearing, consult Cleveland—a company which for 38 years has devoted its engineering skill and production facilities exclusively to the manufacture of high grade, precision-built worms and gears. Cleveland offers a complete line of worm gear sets and speed reducers, including the popular fan-cooled Speedaire—to meet the requirements of any machinery or equipment builder or user for this convenient, right-angle drive.

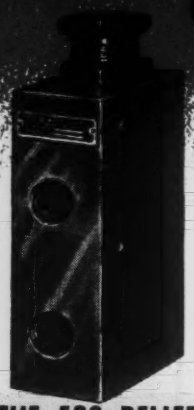
Our sales representatives, located in major industrial market centers, are ready to sit down with you and discuss your power transmission needs, to help you select correct types and sizes. Our service is prompt.

Write for our latest catalog and give us an idea of your requirements. Your inquiry will receive our careful attention.

The Cleveland Worm & Gear Company, 3265 East 80th Street, Cleveland 4, Ohio.

Affiliate: The Farval Corporation, Centralized Systems of Lubrication. In Canada: Peacock Brothers Limited.

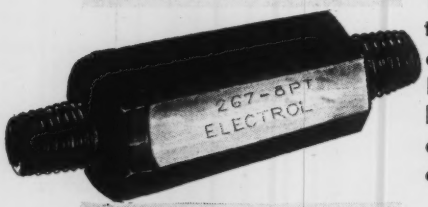
CLEVELAND
Worm Gear
Speed Reducers



THE 523 RELIEF VALVE is compact in design, light in weight, neat in appearance, and can be easily and quickly installed.

No matter whether they're employed to meet the exacting requirements of a maker of X-Ray equipment, or measure-up to the heavy-duty demands of a machine tool builder — Electrol Relief and Check Valves provide precise pressure and flow control . . . *and work quietly together.*

The Electrol Relief Valve is of poppet-type construction, which assures fast, accurate, squeal-free pressure control. It has a wide range of adjustment, and can be set to the desired position, easily and quickly, by turning a fine, screw-thread adjusting cap.



THE 267 CHECK VALVE is for use in air or oil at operating pressures up to 1,500 p.s.i. It is of aluminum alloy construction, and is designed especially for small space installations.

Electrol Check Valves are designed to eliminate any possibility of excessive pressure drop, and are equipped with a patented O-ring check seat, which provides a quick acting, positive check of reverse flow from 0 up to the maximum operating pressure.

Let us supply you with further details, or —if you wish—have one of our engineers call at your convenience.

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Better Designed Products Use Electrol Hydraulics

Electrol

89 GRAND STREET INCORPORATED
KINGSTON, NEW YORK

CYLINDERS • SELECTOR VALVES • FOLLOW-UP VALVES
CHECK VALVES • RELIEF VALVES • HAND PUMPS
POWERPAKS • LANDING GEAR OLEOS • SOLENOID
VALVES • ON-OFF VALVES • SERVO CYLINDERS • TRANSFER
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FOR BETTER HYDRAULIC DEVICES

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Affiliate: The Farval Corporation, Centralized Systems of Lubrication. In Canada: Peacock Brothers Limited.

CLEVELAND
Worm Gear
Speed Reducers

NEED COPIES FAST?

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Copy your drawings, records and reports *fast* with a Bruning Whiteprinter. It's the *fastest, cleanest, most economical* way to copy. You merely place your original on a sheet of Bruning Whiteprint paper and insert both into the Whiteprinter.* Within seconds, out comes an exact, errorproof, smudgeproof copy – a crisp, easy-to-read, properly developed *positive* print!

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You can make an unlimited quantity of prints – any size up to 54" wide by any length – in black-on-white or any of 20 color combinations. They come out flat, dry, stacked. The last copy is as clean and sharp as the first.

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MEETINGS AND EXPOSITIONS

Sept. 18-22—

American Society of Mechanical Engineers. Instruments and Regulators Division Conference to be held at the Municipal Auditorium, Buffalo, N. Y. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Sept. 18-22—

Instrument Society of America. The fifth national conference and exhibit to be held in the Memorial Auditorium, Buffalo, N. Y. Richard Rimbach, 1117 Wolfendale St., Pittsburgh 12, Pa., is executive secretary.

Sept. 19-21—

American Society of Mechanical Engineers. Fall meeting to be held at the Hotel Sheraton, Worcester, Mass. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Sept. 23-26—

Packaging Machinery Manufacturers Institute. Eighteenth annual meeting to be held at the Homestead, Hot Springs, Va. Additional information may be obtained from Society headquarters, 342 Madison Ave., New York 17, N. Y.

Sept. 25-27—

National Electronics Conference. Sixth annual conference and exhibition to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from Kipling Adams, Chairman, National Electronics Conference Exhibit Committee, 920 S. Michigan Ave., Chicago 5, Ill.

Sept. 25-27—

American Society of Mechanical Engineers. Petroleum mechanical engineering conference to be held at the Hotel Roosevelt, New Orleans, La. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Sept. 26-29—

Association of Iron and Steel Engineers. Annual iron and steel exposition to be held at the Cleveland Public Auditorium, Cleveland, Ohio. Additional information may be obtained from Society headquarters, 1010 Empire Bldg., Pittsburgh 22, Pa.

Oct. 12-13—

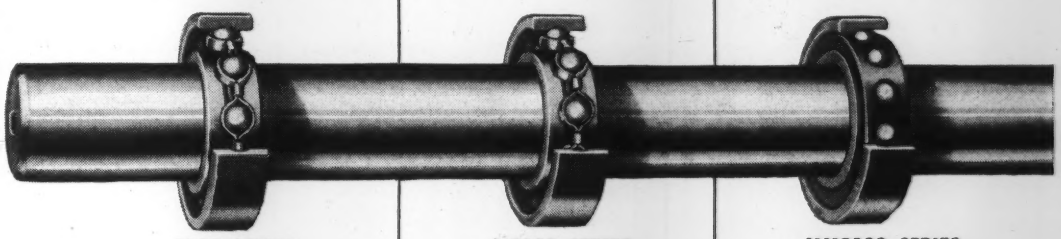
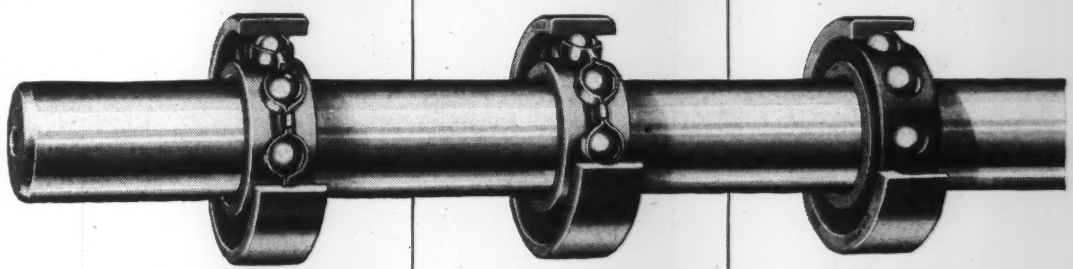
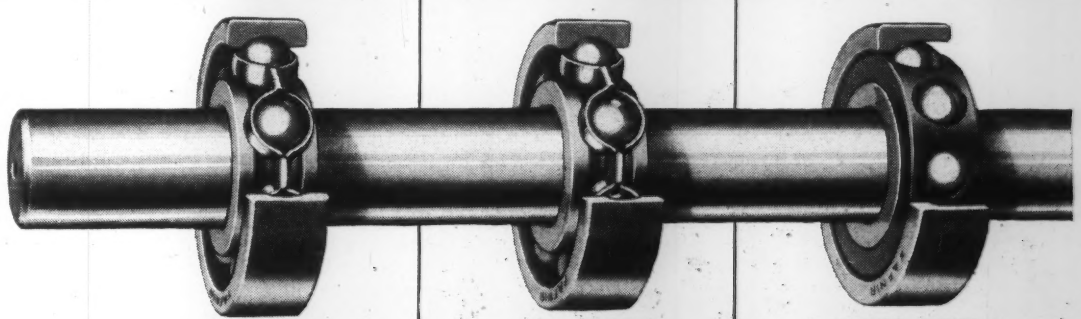
Gray Iron Founders' Society. The

Which for YOUR MONEY?

ball bearings that just "get by" or the best ball bearing "buy"

THIS CHART PROVES IT! You can specify the shaft size, speed, load, service conditions, space and weight limitations . . . and get the Fafnir Ball Bearing that meets those specifications exactly. No need to take less ball bearing or more than just what you need . . . because Fafnir makes them all . . . for all industries.

You get something else, too, that makes Fafnir Ball Bearings a particularly good buy. It's an attitude and an aptitude, a way of looking at ball bearings from where you're sitting and an aptitude for doing just what you'd like done about it, gained from working with not just one or two industries but with all industries.

	Standard	Special Precision	Super Precision
EXTRA-LIGHT **9100 Series** Compact, space-saving, large bore, light cross section design.	 9100 SERIES	 M9100 SERIES	 MM9100 SERIES
LIGHT **200 Series** for loads relatively light in proportion to shaft capacity.	 200 SERIES	 M200 SERIES	 MM200 SERIES
MEDIUM **300 Series** for medium or average loads and continuous operation.	 300 SERIES	 M300 SERIES	 MM300 SERIES

Fafnir Radial Ball Bearings are available in standard shaft sizes in single and double row designs, in duplex arrangements of matched pairs (tandem, back-to-back and face-to-face), and with various arrangements of seals and shields. Fafnir also makes a complete line of ball and roller bearing power transmission units. Write for catalog. The Fafnir Bearing Company, New Britain, Conn.

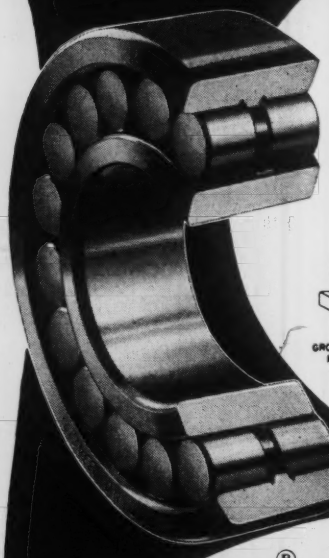
FAFNIR

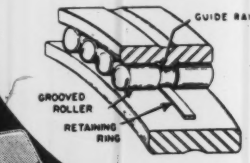
BALL BEARINGS

MOST COMPLETE LINE IN AMERICA

another FIRST in McGILL precision bearings

a NEW PRINCIPLE in roller bearing design

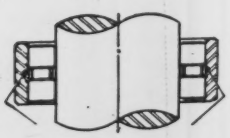




Rollers cannot skew. A guide rail integral with the outer race maintains alignment of the grooved rollers.

The NEW McGILL® GUIDEROL roller bearing

The Bearing with the Self-Aligning Rollers — a simplified roller bearing construction that automatically corrects roller skewing — without space wasting cages! Perfect roller alignment with resulting superior performance is maintained by this new principle of guided rollers. Even with reduced internal radial looseness a GUIDEROL bearing overcomes any tendency to bind under limited misalignment. Produced in needle and cylindrical type roller bearings. Write for information packed Bulletin today.



UNIVERSAL MOUNTING

Universal Mounting — from horizontal to vertical is possible. No cocking of rollers on vertical shafts because the grooved rollers are located by guide rail.

OPEN ENDS — SELF CLEANING

Open Ends — Self cleaning in applications permitting free circulation of oil or grease. Open faces set up flushing action inside the bearing.

GREATER ANGULAR RIGIDITY

Greater Angular Rigidity is provided by longer rollers that permit race and roller contact virtually to the outside edges of the races.

Write for your copy of the new GUIDEROL Bearing Bulletin No. GR-50. Address: McGill Manufacturing Co., Inc., Bearing Division, 200 N. Lafayette Street, Valparaiso, Indiana.

McGILL® precision bearings

22nd annual meeting to be held at the Netherland-Plaza Hotel, Cincinnati, Ohio. R. L. Collier, 210 National City Bldg., Cleveland 14, Ohio, is executive vice president.

Oct. 16-18—

Society of Automotive Engineers. Transportation meeting to be held at the Statler Hotel, New York, N. Y. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Oct. 16-20—

National Safety Council. The thirty-eighth congress and exposition of the National Safety Council will be held at the Stevens, Congress and Morrison Hotels. Additional information may be obtained from R. L. Forney, General Secretary, 425 N. Michigan Ave., Chicago 11, Ill.

Oct. 18-19—

National Conference on Industrial Hydraulics. Sixth annual meeting to be held at the Sherman Hotel, Chicago, Ill., under the sponsorship of the Illinois Institute of Technology and local sections of ASCE, ASME, SAE, Western Society of Engineers, ASLE, AIChE and IAS. Frank W. Edwards, Illinois Institute of Technology, 3300 South Federal St., Chicago 16, Ill., is conference secretary.

Oct. 18-20—

Society of the Plastics Industry. Annual national conference to be held at New Ocean House, Swampscott, Mass. William T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

Oct. 23-25—

American Gear Manufacturers Association. Semi-annual meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Newbold C. Goin, Empire Bldg., Pittsburgh, Pa., is executive secretary.

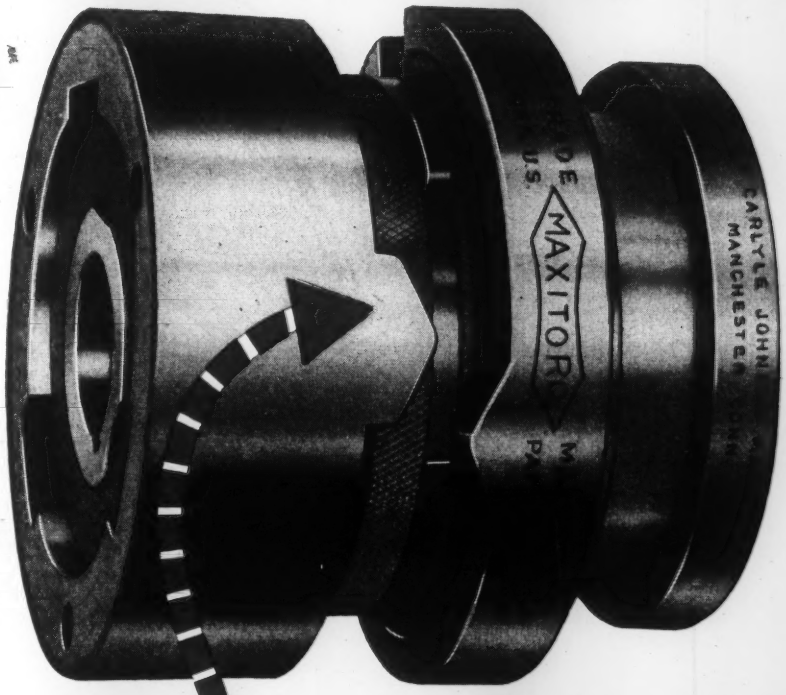
Oct. 23-27—

American Institute of Electrical Engineers. Fall meeting to be held at the Skirvin Hotel, Oklahoma City, Oklahoma. H. H. Henline, 33 West 39th St., New York 18, N. Y., is secretary.

Oct. 23-27—

National Metal Congress and Exposition. The 32nd annual metal congress and exposition to be held at the Chicago International Amphitheatre, Chicago, Ill. W. H. Eisenman, 7301 Euclid Ave., Cleveland, Ohio, is managing director of the exposition.

PROTECT MACHINE AND PRODUCT WITH MAXITORQ



Automatic

OVERLOAD *RELEASE* CLUTCH

When an accidental overload occurs on some types of today's high speed automatic machinery ... packaging, can and bottle labeling machines, for instance . . . confusion reigns. Bottles may be broken and jammed, labels spoiled, machine mechanisms damaged . . . perhaps operator injured. And expensive down-time bites into profits.

But . . . not if your power transmission is protected by the Maxitorq automatic Overload Release Clutch. When overload is a heavy shock or is substantially greater than normal driving load, the clutch instantly and automatically releases. When jammed condition is cleared, the clutch is re-engaged and operation continues. A simple finger-tip adjustment sets the clutch to transmit the normal running load.

OVERLOAD RELEASE TYPE CUPS are the means for connecting the clutch to the driven or driving member. Hub, Ring, Flange or Cut-off Coupling types are available to suit specific conditions.

Maxitorq "Floating discs" actually ride free in neutral . . . preventing drag, abrasion and heat. There are 6 standard clutch sizes with capacities from ¼ to 5 H.P. at 100 r.p.m. Types are wet or dry, single or double, the latter on special order.

The Overload Release Clutch *does* have definite limitations . . . it must be properly applied for maximum effect. Maxitorq engineers, experienced in its varied applications, will gladly provide practical recommendations. Complete specifications on request.

Send for NEW Bulletin No. MD 8

THE CARLYLE JOHNSON MACHINE COMPANY
MANCHESTER • CONNECTICUT

ITS LIFETIME OF
HARD KNOCKS
puts a materials testing laboratory right at your fingertips.

A bowling ball is made to specifications that are *tough!* Tougher than most machine parts.

It takes hard knocks with a bounce, and at the same time retains its perfect balance and smooth finish.

In the heat of a game it is subjected to corrosive perspiration, floor wax, water, chalk, and even cleaning detergents. Yet a genuine Ace Hard Rubber Bowling Ball remains faithful to its owner for years.

That's because it's made of tough Ace Hard Rubber, with tensile strength of 9,000 psi, flexural strength of 11,500 psi. And it's the same impervious material that's used for piping acids.

It's a good example of fabricating, too, with amazingly close tolerances on weight and diameter. It starts as a molding. Then it's ground and polished to give that smooth, satiny feel. Finally the finger holes are machine cut.

Yes, it shows some of the reasons why Ace Hard Rubber is preferred for thousands of parts for machines, appliances, automobiles, furniture, etc. Chances are it's best for some of *your* parts, too. Why not give it a whirl?

Send for free 60-page Ace Handbook
—a gold mine of helpful data

ACE
HARD RUBBER and PLASTICS
AMERICAN HARD RUBBER COMPANY
Since 1851
11 MERCER STREET • NEW YORK 13, N. Y.

DESIGN ABSTRACTS

Electric Induction Drives

THE electric induction coupling lends itself to extremely fine control in smoothness of pickup, control of torque, and control of output speed.

Mechanical and electrical engineers are continually faced with problems of transmitting rotating power in all branches of industry where it is necessary to control either the speed or the torque transmitted. The prevailing power supply of the country consists of alternating current of fixed frequency, which means that motors driven thereby run at corresponding speeds. A number of methods for obtaining speed control are in use today, and all have certain features to recommend them as well as certain disadvantages. An easily and commonly understood method of varying speed is by means of a constant speed dc generator feeding current to an adjustable speed dc motor. For many years this was practically the only method commonly used, and under this condition the question of efficiency in connection with such a system is seldom considered.

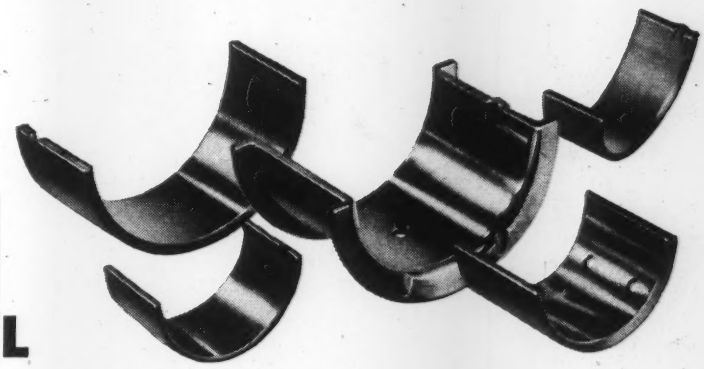
When new methods are proposed, as the electric induction drive, the question of efficiency is invariably encountered, particularly because economic factors are always of great importance. Efficiency of the electric industion drive is about on a par with other systems attaining the same end.

Any device which transmits power from one rotating shaft to another where there is a loss of speed between input and output shafts will generate heat in proportion to torque transmitted, and the difference in speed between the two shafts.

For the purpose of discussion, let us assume that the input shaft delivers 100 hp to the driving member at any desired speed. Assume that the requirements of the driven member for speed are such that the reduction will be 25 per cent below the input speed. The heat loss between the driving and driven member will be 25 per cent of the horsepower or 25 hp. In the case of the Dynamatic induction drive, this loss is nearly all represented by heat generated in the inductor member; therefore, this

✓LONGER BEARING LIFE

✓HIGHER PERFORMANCE STANDARDS



FEDERAL-MOGUL *Sintered* COPPER ALLOYS

Investigate the advantages of superior bearing performance available through our exclusive method of sintering copper lead alloys.

In this process each tiny particle of powder is a true alloy of the desired proportions of copper and lead. In the finished bearing this process has created a known and controlled grain structure, of exceptional purity, with excellent lead distribution and with no segregation. Its ductility and strength are superior to cast materials of comparable analyses.

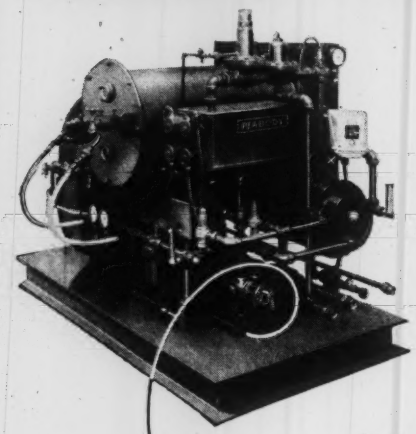
A request on your letterhead will bring a copy of our illustrated technical bulletin describing the development of this improved alloy made possible through powder metallurgy. We believe you will find it a worthwhile addition to your reference file.

FEDERAL-MOGUL CORPORATION, 11045 Shoemaker, DETROIT 13, MICH.

FEDERAL Mogul
Silent SLEEVE BEARINGS

Over 50 Years of Continuous Bearing Experience

ROPER SERIES F PUMP



INTEGRAL PART OF THE

PEABODY

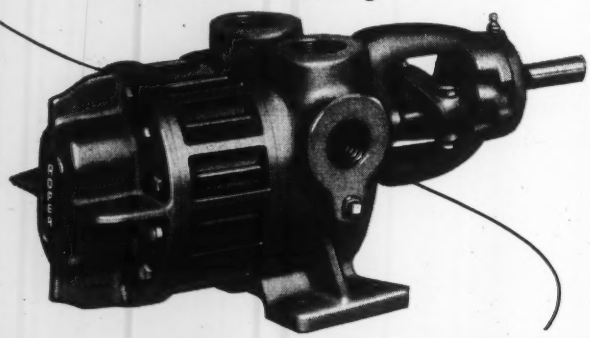
AUTOMATIC BURNER

★

Here is what a Peabody catalog says about the Roper

"Quiet, Slow Speed Operation — The pump used in the Peabody Automatic Burner operates at relatively low speeds. As a result, excessive noise and operating maintenance usually associated with high-speed pumps is entirely eliminated, as well as excessive maintenance costs from worn out parts."

Pressures to-
300 P.S.I.
Capacities-
1-300 G.P.M.



QUIET, DEPENDABLE OPERATION...4-PORT DESIGN

Bunker "C" oil or heavy fuels used in industrial burners must be supplied at the *exact pressure* and *exact viscosity* to get maximum burner efficiency. That is one of the many reasons why Peabody installs the Roper Series F Pump as original equipment on their units.

This smooth-operating, dependable pump is easily installed in this and similar applications. Saves work, time, labor costs. Four-port design makes this possible. This offers eight optional piping arrangements — 4 for CW and 4 for CCW rotation. If you require a pump in sizes 1–300 G.P.M., pressures up to 300 P.S.I. with parts and construction features that are service-proved for long-life, investigate the Roper Series F. Standard fitted and special fitted models to suit specific needs.

Send For Literature Today

GEO. D. ROPER CORP.
248 Blackhawk Park Ave.
ROCKFORD, ILLINOIS

ROPER *Rotary Pumps*

DEPENDABILITY SINCE 1857

member must be adequately fitted with radiating surface sufficient to dissipate that amount of power in heat. Two methods are used. Air cooling is employed, requiring extended surfaces from the inductor drum in the form of fins made of either steel, copper, or aluminum. Liquid cooling is easily accomplished with either oil or water flowing across the inductor to remove the heat generated by the speed reduction represented by slip. Where oil cooling is used, heat exchangers are required to remove the heat from the oil. In many cases water cooling requires so little water that it is most economical to let the water be wasted.

Serves Also As Brake

It is obvious that this type of power transmitter can also be used as a power absorber such as a brake. If one member is held stationary and the other rotated, it is obvious that the torque delivered between the two members is used to brake the revolving member.

Torque delivered by the drive is a function of the total magnetic flux and the rate of slip between the surfaces of the field and the inductor drum. When there is no difference in speed between the two members, no torque is generated as there is obviously no movement of flux lines through the inductor drum. As the relative speed between the two members increases from zero, the torque developed increases with extreme rapidity until the maximum is obtained, after which the torque levels off and increased slip makes no appreciable change.

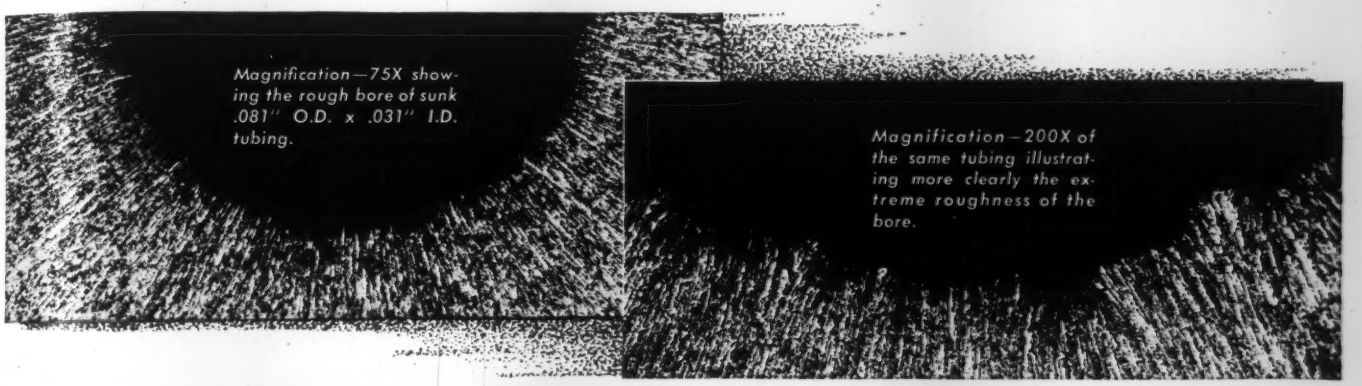
Speed control is accomplished in various ways, but the most common is that of controlling excitation to the magnetized member of the coupling through the medium of electronic tubes. An ac current supply is fed to a small power pack comprising grid-controlled rectifier tubes. The grid of the control tube is caused to add or subtract current to the magnetizing coil with a slight variation in speed of the output shaft of the coupling. A small permanent magnet ac generator attached to the output shaft of the coupling delivers the signal to the electronic tubes for controlling the speed. A small simple rheostat, placed at the operator's station or on the electronic control panel, adjusts the governing voltage. Speed of the output shaft is held to within relatively few revolutions per minute of the selected speed over a large power variation.

Where it is desirable to manually control the speed, a simple rheostat

WHICH TUBE WOULD YOU CHOOSE?

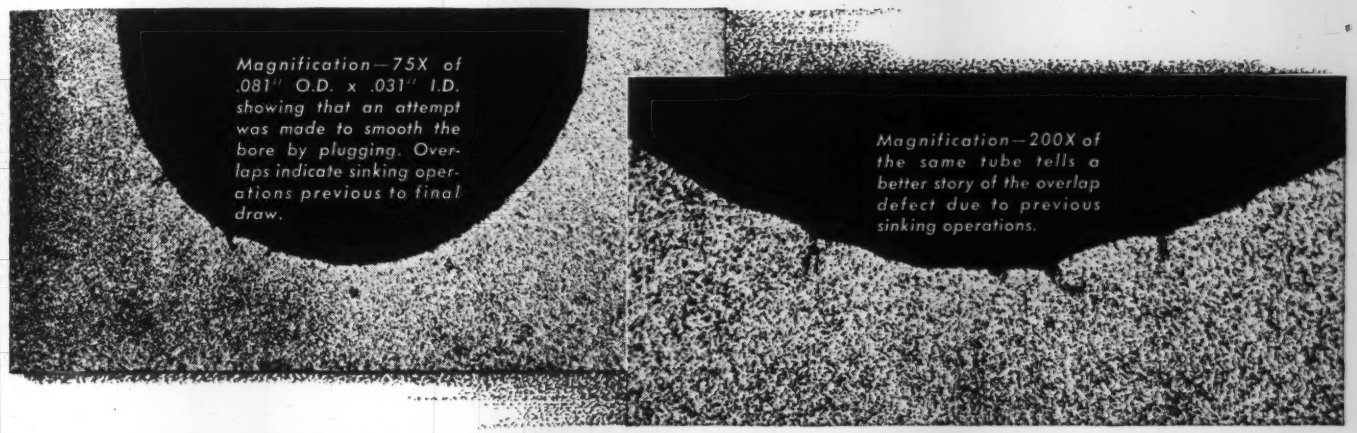
These photomicrographs show the transverse sections of three brands of capillary tubing.

They tell their own stories and prove the truth of our statement that "There IS a difference in tubing."



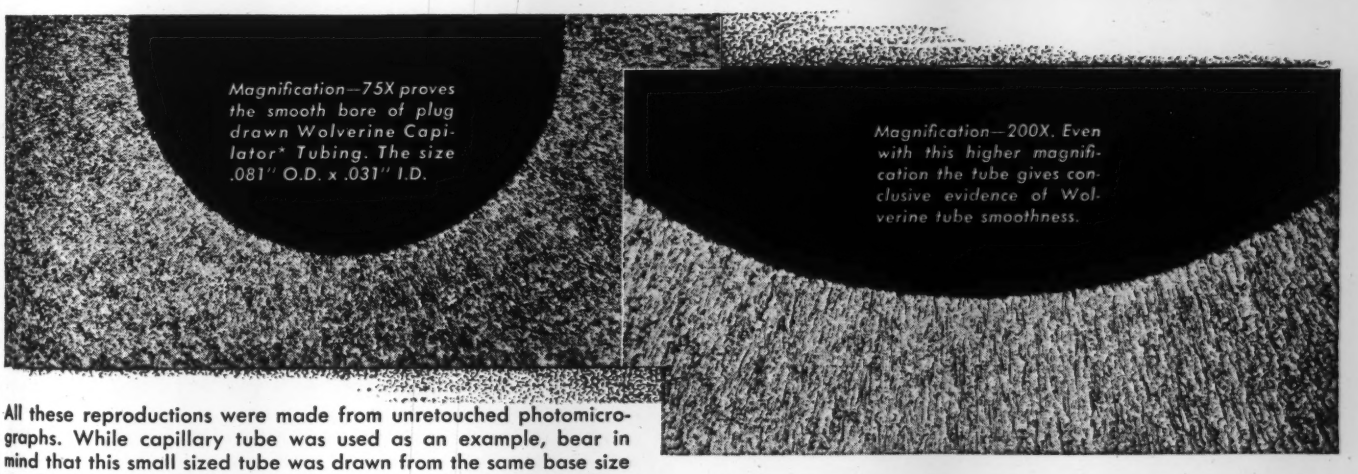
Magnification—75X showing the rough bore of sunk .081" O.D. x .031" I.D. tubing.

Magnification—200X of the same tubing illustrating more clearly the extreme roughness of the bore.



Magnification—75X of .081" O.D. x .031" I.D. showing that an attempt was made to smooth the bore by plugging. Overlaps indicate sinking operations previous to final draw.

Magnification—200X of the same tube tells a better story of the overlap defect due to previous sinking operations.



Magnification—75X proves the smooth bore of plug drawn Wolverine Capilator Tubing. The size .081" O.D. x .031" I.D.*

Magnification—200X. Even with this higher magnification the tube gives conclusive evidence of Wolverine tube smoothness.

All these reproductions were made from unretouched photomicrographs. While capillary tube was used as an example, bear in mind that this small sized tube was drawn from the same base size (3" O.D.) used to produce all other sizes of Wolverine tube. Thus this evidence of quality applies to ALL Wolverine tube, regardless of size.

Which tube would you choose? Of course, you'd choose the tube which would allow the smoothest flow of liquid or gas—and, in the long run, give you the most economical service. You'd choose dependable Wolverine tube.

*Reg. U.S. Patent Office

Let us send you literature telling about the use of tubing in your own applications.

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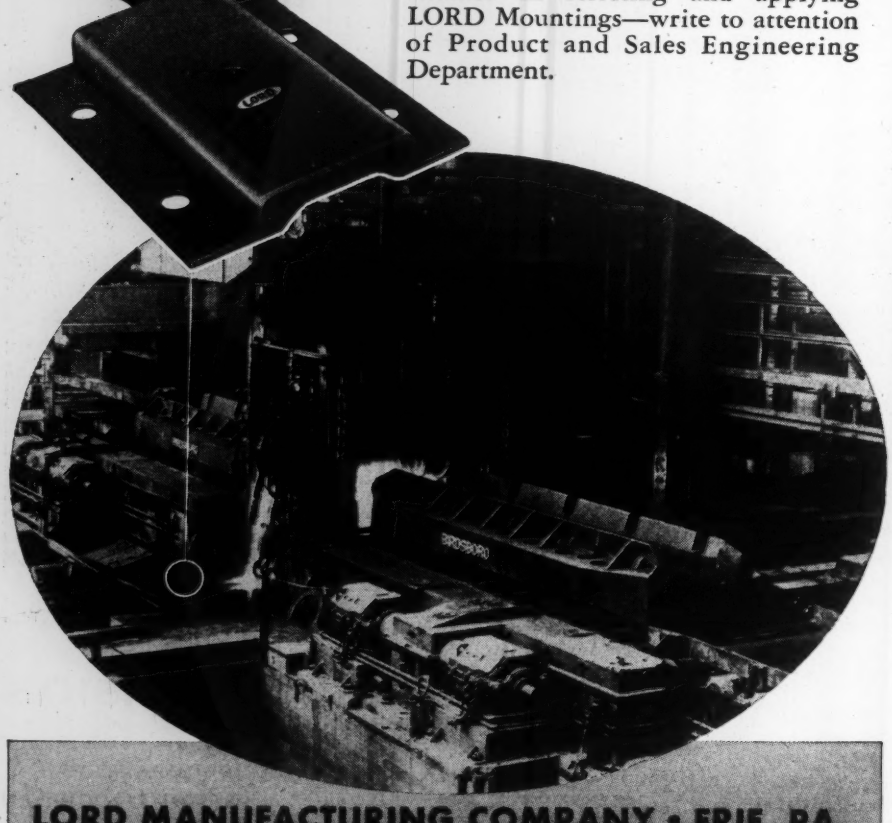
in HEAVY INDUSTRY, too, LORD MOUNTINGS

Furnish Shock Protection

This Birdsboro Blooming Mill illustrates how LORD Mountings effect economies in machine design, and how they protect working parts against damage from shock loads. Steel ingots and slabs are manipulated on the table at left of mill. They drop several inches during turning and rolling operations, creating heavy shock loads on table rollers and bearings.

By incorporating LORD Industrial Shockmounts into the design, Birdsboro engineers were able to greatly increase the capacity of an existing table design with a moderate increase in cost, where otherwise a very heavy table at considerable additional cost would be required. A faster acting table was obtained . . . bearing and roller stresses were reduced . . . service life was lengthened . . . maintenance was lowered.

LORD Mountings improve product performance . . . make machines operate smoothly and quietly . . . add sales appeal and customer satisfaction . . . often reduce manufacturing and assembly costs. Designers should investigate the advantage of LORD Mountings for both heavy and light equipment. For further information—or for assistance in selecting and applying LORD Mountings—write to attention of Product and Sales Engineering Department.



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LORD BONDED RUBBER

Vibration-Control Mountings . . . Bonded-Rubber Parts

is used in connection with a supply of direct current to the magnetizing coil.

Where torque is required to be controlled rather than speed, the same type of electronic system for excitation described in the foregoing is used. Instead of the speed control signal being derived from rotation of the output shaft, the signal for controlling torque is obtained from a small current transformer introduced into the supply line to the ac motor driving the coupling. For torque control of couplings, not connected with electric motors, various means are used including a torque responsive reactor affected by torque at either the input or output member of the coupling. Control of torque can also be had manually by varying the direct excitation by a rheostat.

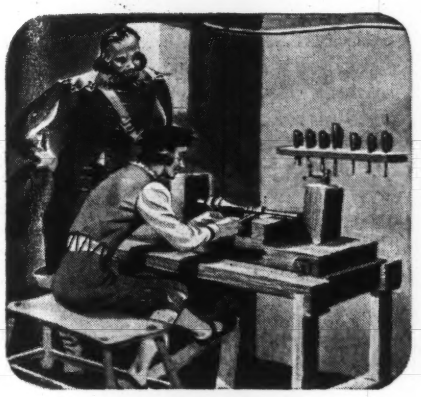
The ability of this type of drive to control speed, to act as a power disconnect, and to absorb torsional vibrations makes it ideal for an unending list of applications.

The inductor principle together with the close control available solved many engineering problems in the development of all types of engines during and since the war. High speed turbines for jet propulsion engines can be tested by directly connecting them to this type of dynamometer.

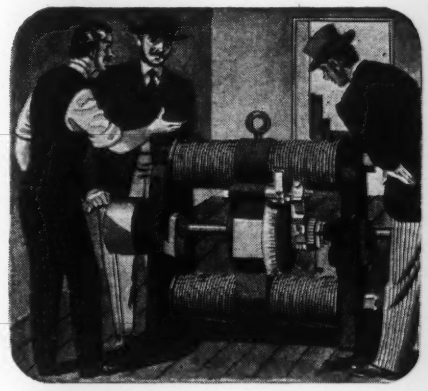
Greater in importance to industry, however, is the application of couplings for industrial use. They are used on heavy-duty power shovels for operating the shovel and auxiliary movements, and are also used on a great number of hoisting apparatus and cranes. *From a paper by Martin P. Winther, vice president, Eaton Manufacturing Co., at the Spring Meeting of the ASME, April, 1950, in Washington.*

Engineer's Role in Community Affairs

IN the present complex and constantly changing national and world order there can be no excuse for any individual's failing to carry a reasonable share of the total community load. This is especially true of professional men and women, those upon whom the best we have in education has been lavished. This "community load" does not consist entirely of what we too frequently deride as politics. Indeed, the bulk of it has either nothing at all to do with politics, or at best is indirectly related to politics. I have in mind all those myriad enterprises in which people come together to accomplish purposes for mutual and public benefits as contrasted with strictly



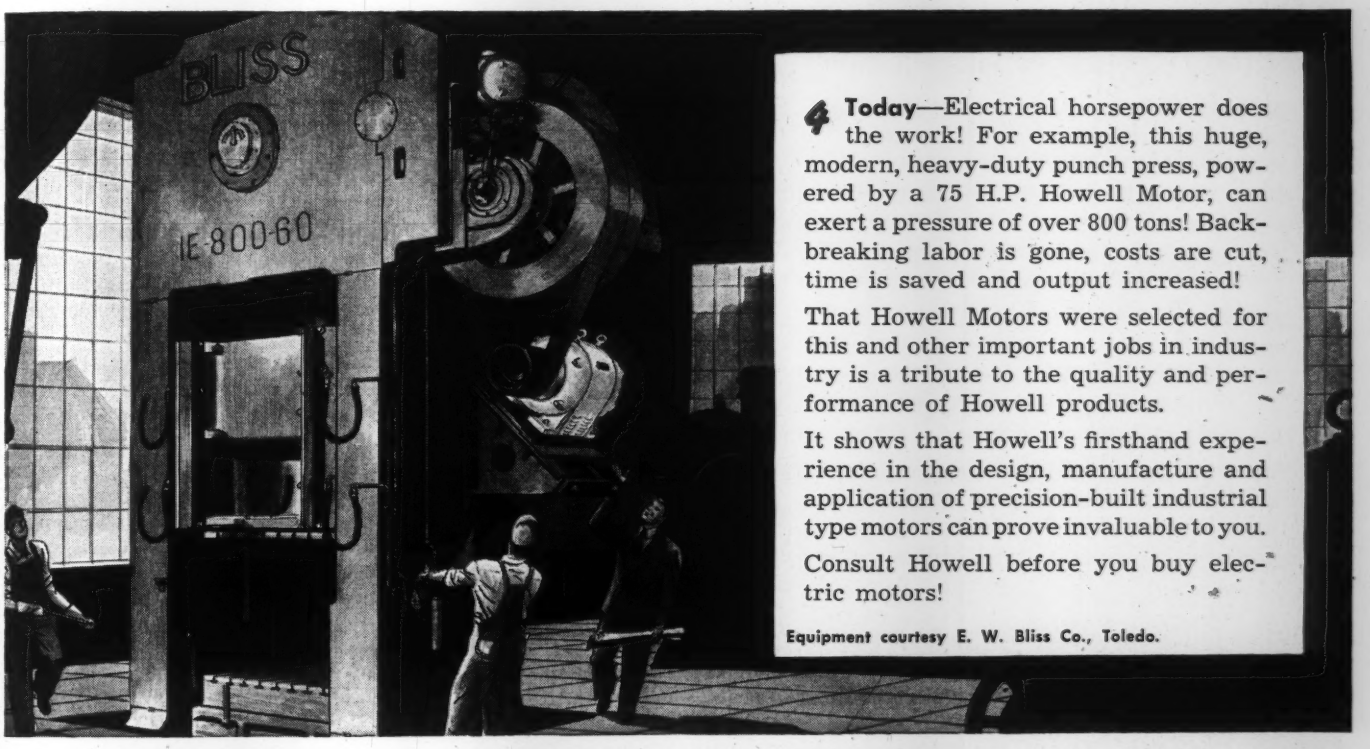
1 1600—Man worked hard for his pay in these days. Crude machines like this began to lighten his load by replacing laborious handwork with foot power. But progress was really only beginning.



2 1875—By accident, it was discovered that if two direct current dynamos were connected together, one would serve as a generator and drive the other as a motor. Electrical horsepower was here!

3 1900—Result: Electrical horsepower replaced foot power! By 1915, Howell "Red Band" Electric Motors arrived. Soon, these rugged, industrial type motors were widely accepted in many industries.

NOW...A PUNCH PRESS WITH A PUNCH!



4 Today—Electrical horsepower does the work! For example, this huge, modern, heavy-duty punch press, powered by a 75 H.P. Howell Motor, can exert a pressure of over 800 tons! Back-breaking labor is gone, costs are cut, time is saved and output increased!

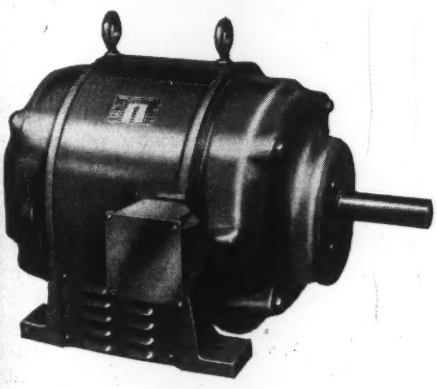
That Howell Motors were selected for this and other important jobs in industry is a tribute to the quality and performance of Howell products.

It shows that Howell's firsthand experience in the design, manufacture and application of precision-built industrial type motors can prove invaluable to you.

Consult Howell before you buy electric motors!

Equipment courtesy E. W. Bliss Co., Toledo.

Free enterprise encourages mass production, supplies more jobs—provides more goods for more people at less cost.



Howell Type F Punch Press Motor

HOWELL MOTORS

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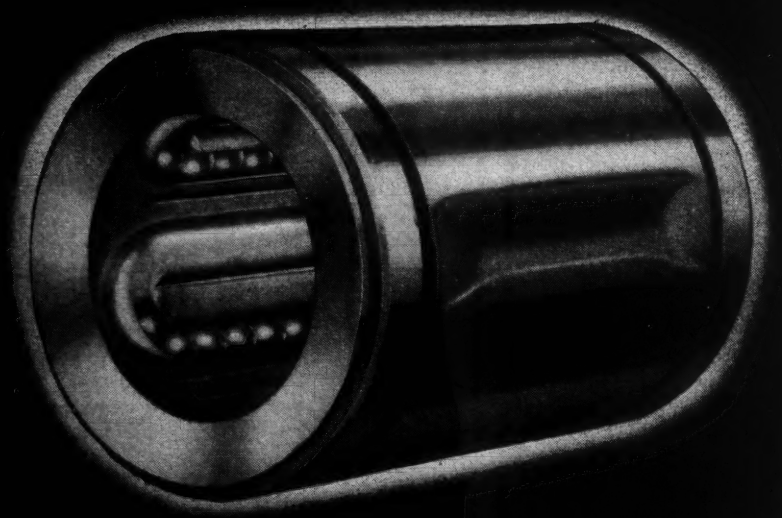
HOWELL Red Band MOTORS

Announcing A New LOW COST Ball Bearing for Your

LINEAR MOTIONS



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THE COMMERCIAL GRADE SERIES B

Sliding linear motions are nearly always troublesome. Thousands of progressive engineers have solved this problem by application of the Precision Series A Ball Bushing.

The low-cost Commercial Grade Series B bearing is now added to the Ball Bushing line and offered to original equipment manufacturers. This ball bearing has been developed for support of linear motions in competitively priced, volume produced products where super precision is not essential. Alert designers can now make tremendous improvements in their products by using Ball Bushings on guide rods, reciprocating shafts, push-pull actions, or for support of any mechanism that is moved or shifted in a straight line.

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- LOW FRICTION
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- SOLVES SLIDING LUBRICATION PROBLEMS
- LASTING ALIGNMENT
- LOW MAINTENANCE
- LONG LIFE

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Now in production for ¼", ½", ¾", 1" and 1½" shaft diameters. If you have a catalog, phone your representative or write us for new Data Sheet. If not, write for complete literature and the name of our representative in your city.

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PROGRESSIVE MANUFACTURERS USE BALL BUSHINGS— A MAJOR IMPROVEMENT AT A MINOR COST

individual, gains. They may have social, cultural, or humanitarian objectives, or a combination of all three.

Failure of the engineer and scientist to participate in community affairs has been established statistically. The figures give evidence that among several hundred thousand individuals in the United States, educated in one way or another to practice engineering, only a negligible percentage interest themselves in public affairs.

Distaste for Politics No Excuse

We engineers seem to utilize our distaste for politics as an excuse for laying off on all types of community service. What we are prone to denounce as politics is of course only one phase of community life, even if it is one of the most important. Through its local, state, national and international constituent parts, government—or politics—is the structure by which our whole civilization is supported. It is impossible to conceive of either science or engineering without this bedrock of human society. Without government there could be no engineering as we practice it. Do not let us fool ourselves. We do not in reality stand aloof from government, no matter how we try. On the contrary, the closer we can make the tie-in between government and engineering, the greater will be the profession's opportunity for service.

Our dislike for war does not keep us from accepting our respective assignments when war becomes a necessity. So our dislike for the ethically nether side of partisan politics is not a sufficient excuse for refraining, as most of us do, from all manner of community service. We engineers must build our own place in society. Qualifying as a reasonably good citizen will, of course, require only a fraction of the time, thought and energy which go into the making of a good engineer. So the frequently used excuse of lack of time has little validity.

There are many ways in which an interested citizen may make his influence felt without entering the political arena or running for office. Jefferson cited as constituting the essence of a republic a synthesis of "action by the citizens in person in affairs within their reach and competence, and in all others, by representatives chosen immediately, and removable by themselves."

It has always seemed to me a mistake to urge adult engineers to enter politics in view of the fact that few of them have had the seasoning fitting them for such an exacting and specialized calling. The best train-

Look at this difference in
MACHINING COSTS

SAE 52100 ball bearing raceway about 2" O.D., ¼" wall and ⅝" wide.

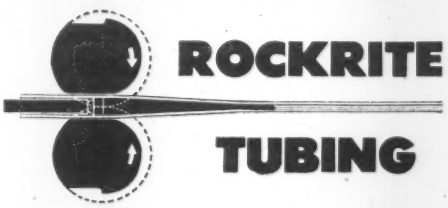
These startling figures were developed from actual cost records. The difference of 1.9¢ per part is the result of two important Rockrite advantages:

SMALLER MACHINING ALLOWANCE — That's because Rockrite Tubing is sized by a distinctively different process to *much closer tolerances* than are practical by any other method.

BETTER MACHINABILITY OF METAL — Because Rockrite Tubing is sized by cold-compression; has a denser, more uniform microstructure, better work surfaces — permitting higher cutting speeds and feeds. In this case machine output is some 60% higher.

The extra cost of Rockrite stock is only six-tenths of a cent per piece, and the costs shown above do not include savings in handling, coolant, rejects or magazine stocking.

Do you know the 3 requirements essential for tube accuracy and cost savings? New Bulletin R2 tells you . . . gives all the facts on close-tolerance Rockrite Tubing. Send for your copy today.



Leads all others in these savings

- **Higher cutting speeds**
- **Tools last longer between grinds**
- **Work-surface finishes are better**
- **Machined parts have closer tolerances**
- **Stations on automatics are often released for additional operations**
- **Extra-long pieces available—less downtime for magazine stocking and fewer scrap ends**
- **Closer tolerances often eliminate necessity for machining on outside or inside**

TR-116

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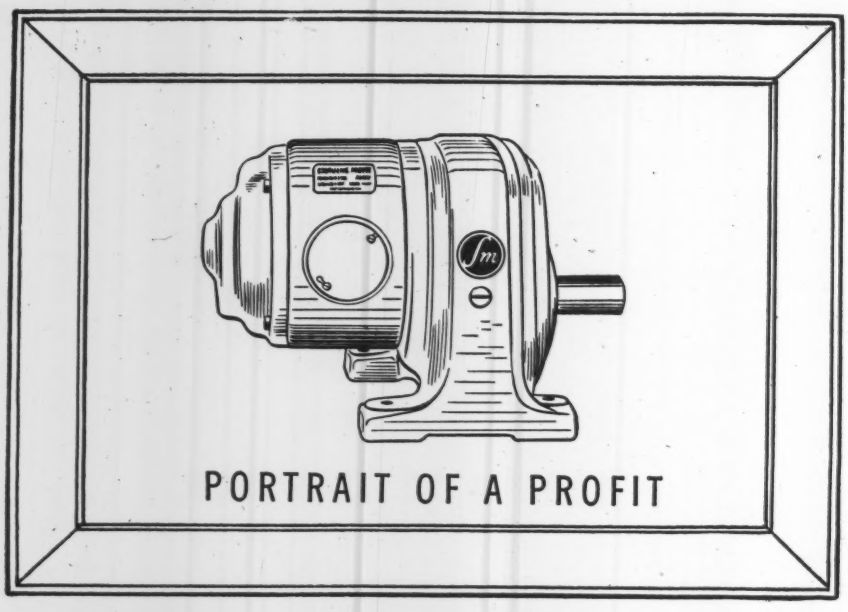

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Write for Bulletin

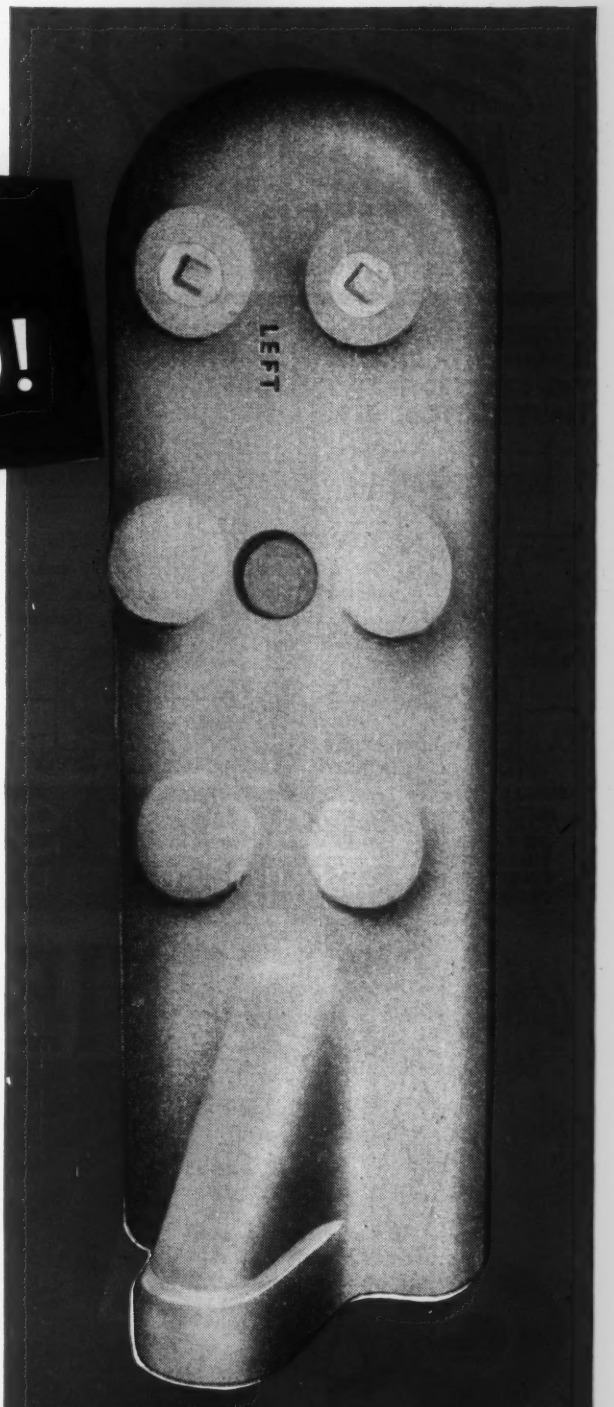
ing for politics is in the less specialized give and take of the garden variety of community affairs. And in such matters we should mix as citizens rather than as engineers. In the primary discussion of community affairs there is no demand for men with professional training. Balanced and sympathetic judgment is wanted whether it comes from the high or low, from the formally educated or uneducated. In the councils of the people we do well to try to forget that we are engineers. If something comes up where our specialized training is useful, let us utilize the opportunity. But we should not wait to be called.

No Place for Hypercritic

Granted that there are quite a limited number of engineers so equipped and conditioned as to be able to serve society well by holding public office, there is little warrant for the quite undiscriminating, and at times illiterate, attitude toward politics and politicians that is generally entertained by our group. If the profession of engineering is to play its full role in the new world already in formation, we must abandon what is now almost our universal hypercritical attitude toward government and the personnel and agencies through which it expresses itself. We are associated, of course, in this attitude by a large sector of the business world. We must learn neither to despise nor even to feel apart from this important segment of our common life. This must never, of course, excuse being a party to the illicit or the venal. Nor should it act as a bar to rating critically the individuals in public service. But it does means establishing at all levels an always available bridge of understanding and co-operation between engineering and politics, two great and constructive agencies of modern society. A great good both to the people at large and to our profession would be effected through such a seemingly logical readjustment of our relations with the political life of the nation.

To carry out the dream of a great democratic state, which is the only possible long-time answer to totalitarianism in its various forms, will require that the ideal of public service be nurtured at every age—through earliest childhood, school and college days, and on through post-graduate and adult life. It should be a continuous process of community service in action.—*From a paper by Morris Llewellyn Cooke, consulting engineer in management, presented at the ASEE Annual Meeting, June 21, 1950, in Seattle, Wash.*

THIS CASTING SAVED $12,000.00!



Why an Electrode Holder, redesigned by National Bearing Division, lasted over 73 times longer

A large Midwest plant was getting 5 days service from furnace electrode holders . . . before a trial order was placed with National Bearing Division. This plant saved dollars—*right from the start.* A special copper alloy permitted sound, dense castings at no sacrifice in electrical conductivity. Result? *30 days service—instead of 5—before replacement.*

On the next order, National Bearing Division engineers submitted a new design for the clamp—for greater strength and resistance to cracking. The redesigned electrode holders are still in use—after a year of trouble-free operation! They've saved $12,000.00 on replacement costs and furnace maintenance.

Better, longer-lasting non-ferrous parts can save money in your plant or product . . .

National Bearing Division has complete facilities for finding practical, economical solutions to non-ferrous bearing and casting problems. Investigate these complete facilities—whether your problem requires designed engineering service, or production of non-ferrous parts to your own specifications.

If you design, specify or purchase . . . this catalog will interest you

find out how complete facilities for Research, Engineering and Production, can reduce costs of non-ferrous parts in your plant or product. Write for a free copy of this catalog.

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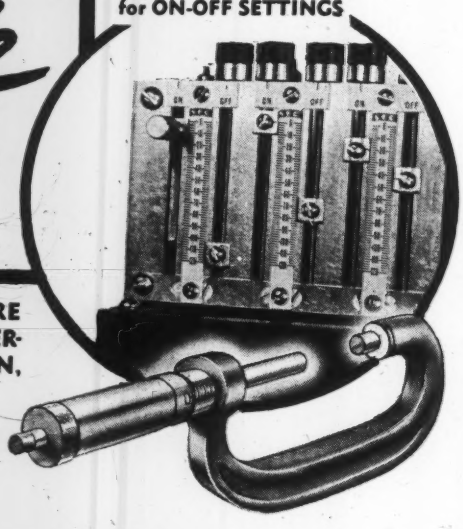
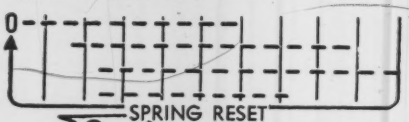
Eagle MULTIFLEX TIMER

MICROMETER KNOBS for ON-OFF SETTINGS



CLOSES OR OPENS ONE OR MORE CIRCUITS DURING A TIME INTERVAL STARTED BY A PUSHBUTTON, LIMIT SWITCH, ETC.



SPRING RESET

Controls 1 to 6 circuits

Used where any electrically-energized devices are to be operated in a timed sequence. Driven by synchronous motor with sealed-in lubrication. Solenoid-operated clutch with spring reset.

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EAGLE SIGNAL CORPORATION Moline Illinois

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MASS production of fine gears is an accomplishment that requires the ultimate in manufacturing facilities and experience. And it's all here at Fairfield—metallurgical departments, batteries of the most modern machines, complete heat treating equipment, all operated by skilled craftsmen producing truly fine gears. Here is 30 years of experience translated into the gear-making "know-how" that is essential to manufacturing high quality, dependable gears.

Our facilities are complete for producing gears for a wide variety of applications. Why not consult Fairfield on your next gear problem? Your inquiry will receive prompt attention.

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NEW MACHINES

And the Companies Behind Them

Manufacturing

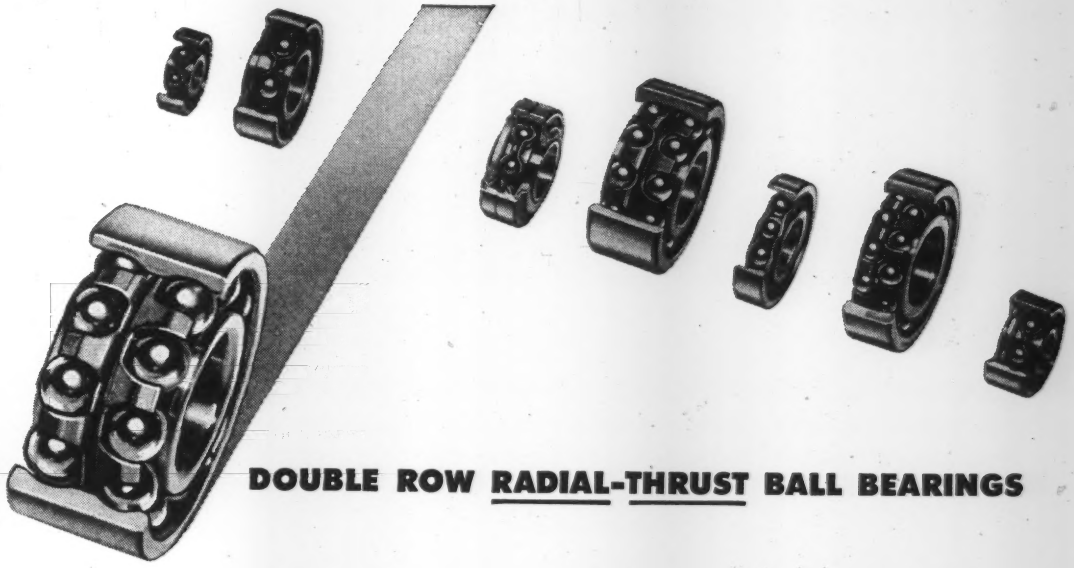
AUTOMATIC INDEX MACHINE: For multiple drilling, reaming, tapping, counterboring or spotfacing operations. Incorporates three Millholland automatic units having 1½-hp capacity and 2½-in. stroke. Part held in air-operated chuck mounted on automatic indexing spindle. Production rate on parts requiring 12 holes around periphery and 24 holes in flange end, 150 to 320 pieces per hour. *W. K. Millholland Machinery Co., Indianapolis, Ind.*

PUNCH PRESS: Hand-operated turret model for piercing, embossing or forming parts in lots of 500 or less. Press has 13-in. throat, weighs 600 lb. Any one of 12 punch and die sets can be brought under ram by moving index handle. Capacity, 10 ton; max punch size, 1½-in. with ⅛-in. mild steel. *Diamond Machine Tool Co., Los Angeles, Calif.*

ARC WELDERS: Transformer type ac units have no moving parts. Electrically adjustable reactor eliminates hand-crank setting of welding current. Model TSP-204-C with power factor correction rated 200 amp at 30 v; Model TSP-182-C with power factor correction and limited input for operation on REA lines rated 180 amp at 25 v. Hand wheel switch controls five main welding current taps. Rheostat control knob regulates dc to reactor to give 100 fine welding current settings for each tap. *Hobart Brothers Co., Troy, O.*

PUNCH PRESS: Capacity, 7½ tons. Shut height (ram up to bed) increased to 9 in. Ram equipped with 1 9/16-in. diameter shank hole. Furnished with standard 1¼-in. stroke with special strokes to 2½-in. available. Bolster plate measures 8 by 11 in.; bed equipped with 4 by 5-in. hole for bottom ejection. Press can be single-tripped or operated continuously. *Benchmaster Manufacturing Co., Los Angeles, Calif.*

LATHE: Model LS semiautomatic low-swing lathe retains features of special machine used for turning gun tubes and high explosive shells. Front turning carriage now equipped with two or more individually-controlled, power-operated, cross slides. Tool cutting pressures taken on rolls in contact with cross feed cam. Includes cam-feed back-



WHEN TO USE DOUBLE ROW RADIAL-THRUST BALL BEARINGS

Heavy thrust or combined loads that have to be engineered into space too small for large diameter single row bearings are applications for compact Federal Double Row *Radial-Thrust* Ball Bearings.

Equal load distribution on each row of balls is insured by accurate alignment of both races—ground in a single setting for absolute concentricity and alignment to each other and to the bore.

Double angular contact takes the heavy radial loads as well as heavy thrust loads in either direction. For maximum rigidity and minimum deflection under all conditions of load, they may be preloaded in assembly if desired.

Our Catalog "K" describes all types of Federal Double Row *Radial-Thrust* Ball Bearings as well as companion series of double row *radial* bearings—shielded and sealed. Ball bearing selection charts among the 100 pages of engineering data will help you compute bearing loads, determine capacities at every speed, and select the type and size best suited to your application—from one of the most complete ball bearing lines. This 260-page book is yours for the asking. Write for a copy today.

Also write for a copy of our latest Ball Bearing Conversion Tables which contain complete up-to-date interchange information.

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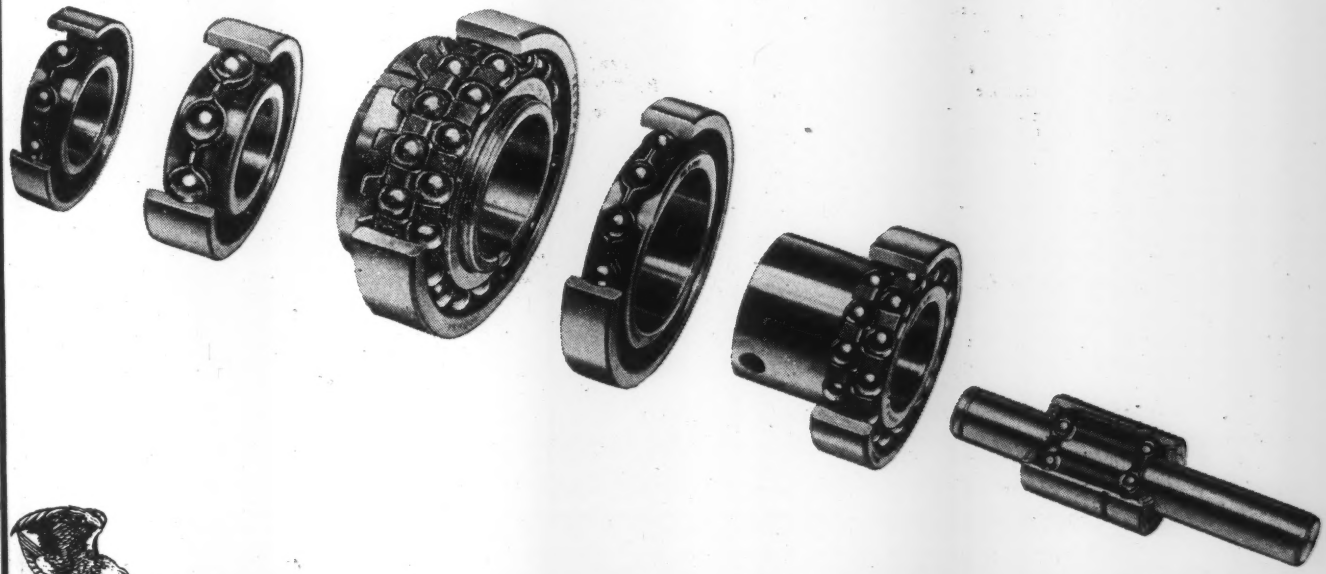

FEDERAL BALL BEARINGS

ONE OF AMERICA'S LEADING BALL BEARING MANUFACTURERS

Quality since 1908

squaring attachments, adjustable stop for positioning carriage in relation to starting point of cut, and automatic tool relief at end of cut. *Seneca Falls Machine Co., Seneca Falls, N. Y.*

PUNCH: Hand-operated Model No. 2 has 12¼-in. throat depth. For short-run blanking, drawing, embossing, and forming operations. Can perforate holes to 4-in. diameter. Force applied to ram through roller bearing cam. Triangular shaped ram cannot twist, maintains accurate position of punch head. Dimensions of bed; 6 by 7½ in. Net weight, 350 lb. *O'Neil-Irwin Manufacturing Co., Lake City, Minn.*

DRILLING AND TAPPING MACHINE: Semiautomatic unit for drilling and tapping radial holes in circular parts such as wheels, rims, etc. Machine includes one Govro-Nelson drilling unit and one tapping unit mounted on table according to number and spacing of required holes. On hand-indexed model, interlock prevents indexing of part until spindles are withdrawn, and vice versa. *Govro-Nelson Co., Detroit, Mich.*

SHEET BUFFING MACHINE: New 48-in. wide openside oscillating type machine for heavy-duty production finishing of ferrous and nonferrous sheets. Has hydraulic table and power elevating and lowering head for 18-in. diameter buff or brush roll. Oscillating mechanism powered by 3-hp motor, has stroke variation from ½ to 2 in., range of 50 to 150 cycles per minute. Main spindle powered by 50 to 100-hp motor. Machine built in table widths to 72 in. and table lengths to 8 ft. *Hill Acme Co., Cleveland, O.*

WELDING TRANSFORMER: Model 400E rated 400 amp at 60 per cent duty cycle, with welding range of 70 to 585 amp. Unit retains full 75 volt open-circuit voltage. Fan motor and capacitors accessible through front of machine. *A. O. Smith Corp., Milwaukee, Wis.*

CORE GRINDER: Uses 48-in. diameter round table mounted on sealed roller bearings. Table rotated manually or by adjustable speed reducer powered by 1-hp motor. Counterbalanced sliding head carries ¾-hp motor powering grinding wheel. Head adjusts vertically. *Standard Pattern Works, Detroit.*

TURRET PUNCH PRESS: for high-speed piercing of electrical panels, radio and television chassis, switchboards, etc. Combines pantograph for rapid hole location with turret for quick access to punches and dies. Turret carries 16 to 20 different punches and dies, handwheel rotates tur-

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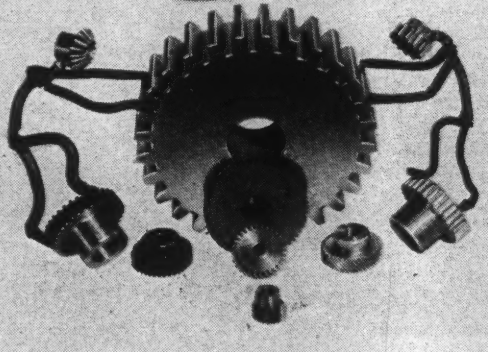

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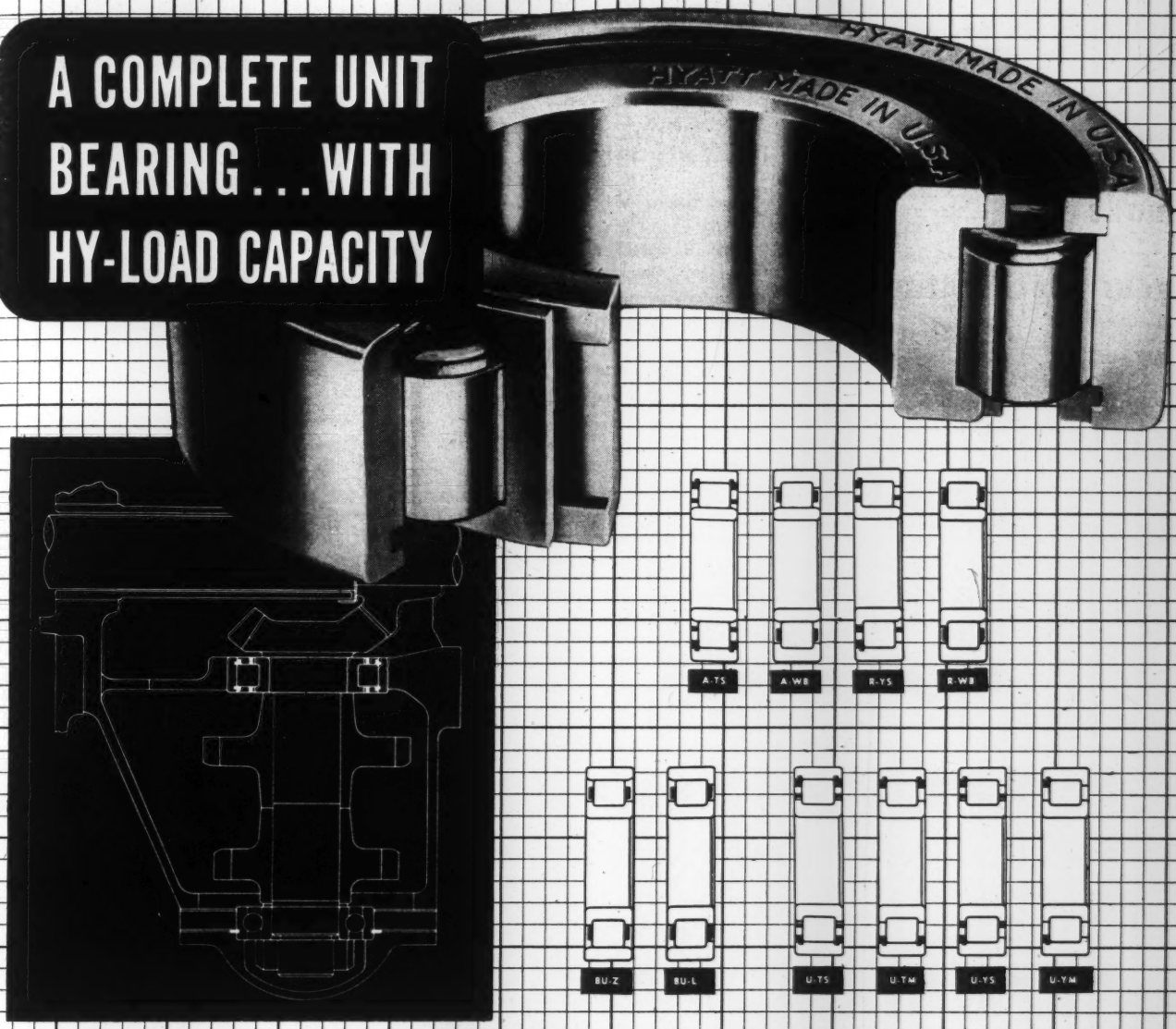

HERE is a cutaway illustration and typical application of the "U-TS" type Hyatt Hy-Load straight cylindrical bearing which has rollers, separator and races assembled into a single non-separable unit. The inner race has flanges and the outer race is grooved at both ends to accommodate retainment rings. The rollers are separated and spaced by a one-piece separator.

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Separable inner race, separable outer race and complete unit Hy-Load Bearings are available with varying combinations of retainment rings and flanges to fit a wide range of application requirements.

• • •

Full information about all Hyatt Hy-Load Roller Bearings is contained in Catalog 547 . . . a complete engineering guide to radial bearing selection and use. Hyatt Bearings Division, General Motors Corporation, Harrison, New Jersey.

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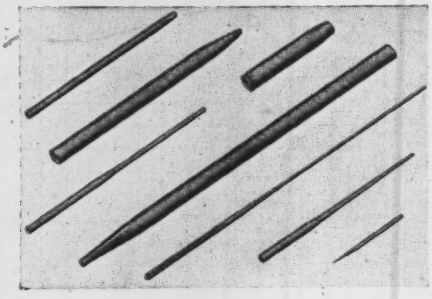

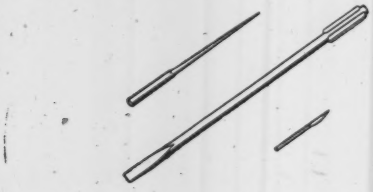

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ret to any punching set in two seconds. Press handles sheets to 28 by 40 in., has 11/16-in. stroke, 175 per minute. Capacity, 15 tons. Unit is powered by 1-hp, 3450-rpm motor. *Wiedemann Machine Co., Philadelphia, Pa.*

LATHE: General purpose Model No. 1 in 12 by 30, 12 by 42 and 12 by 54-in. sizes, all with 14½-in. max swing over ways. Spindle driven by five V belts, with single-lever belt shifter for speed selection from four-step cones mounted on 3-hp reversing motor and on jack shaft. Spindle speed range, 30 to 1142 rpm. Includes 36 selections for threading, from 1½ to 80, and 36 selections for feeds, from 0.0031-in. to 0.1666-in. *Hendey Machine Co., Torrington, Conn.*

ALKYD PRESS: Fully automatic, high-speed unit requires only electrical connection and shop air source. Features of 15-ton hydraulic press include rapid opening and closing speeds of 400 in. per minute and automatic cycling. Automatic feeders standardized in 2, 4, 6, and 8-cavity models, with others available. Test run produced 345 parts per hour from three-cavity setup using alkyd. Material measured and fed into cavities automatically. No preheat or preforming necessary. *Baker Brothers Inc., Toledo, O.*

SPINDLE STAND: For finishing and polishing work to be pushed across table by hand or power feed. Motor and polishing spindle adjustable through 180 degrees, with 20-in. vertical adjustment and 5-in. longitudinal adjustment on spindle. With or without table. *Vonnegut Moulder Corp., Indianapolis, Ind.*

MACHINE VISE: Uses large diameter air cylinder for clamping action. Stroke of standard model, 1¼ in.; 6-in. stroke available for pressing, shaping, etc. Model 811 has 8⅜-in. wide jaws opening to max of 11 in. Jaws open and close quickly without hammer effect. *Vdn Products Co., Erie, Pa.*

BENDING PRESS: Capacity, 300 tons. For blanking clutch rings, clutch gear teeth and special flat blanking operations. Designed for two-man operation. Has air-electric clutch control and four-position stroke speed selector. Both of two sets of control buttons must be depressed to start stroke. Press speed 25 strokes per minute; shut height, 14 in.; throat depth, 19 in.; bed front to back, 3 ft; distance between tie rods, 4 ft. *The Cleveland Crane & Engineering Co., Wickliffe, O.*

TUBE BENDERS: Gear type hand operated units for copper, brass, aluminum or steel tube, including Bundy

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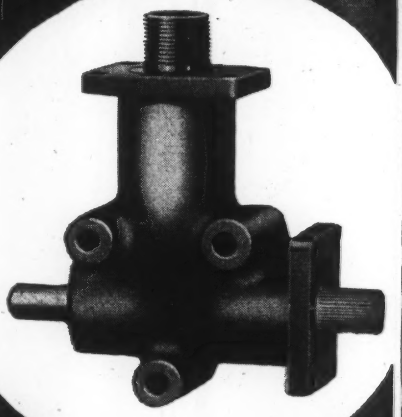

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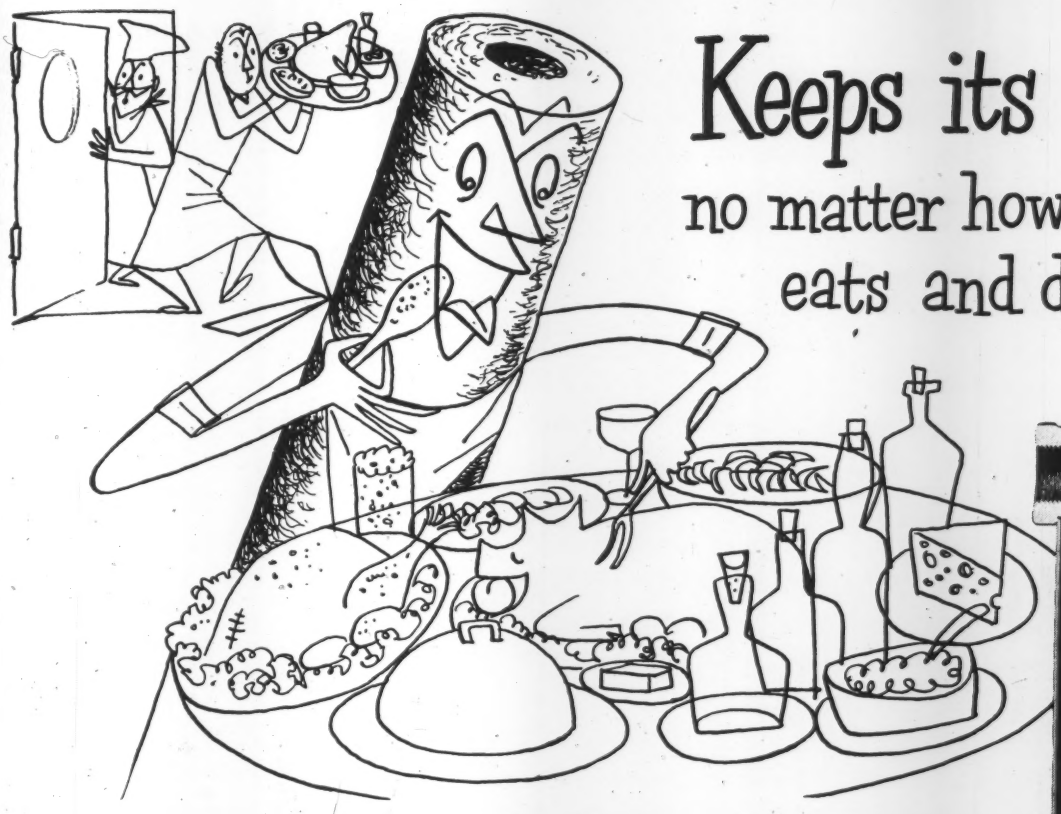

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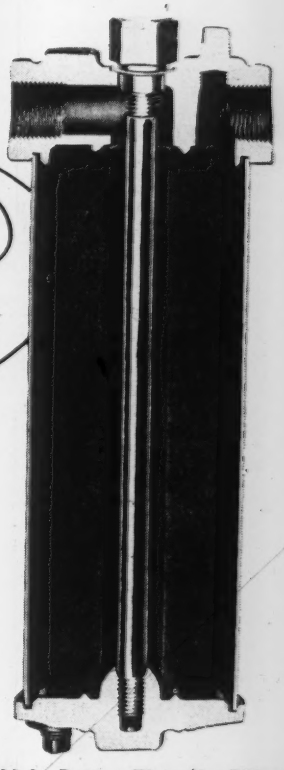

Put the Cuno MICRO-KLEAN filter on almost any kind of a diet . . . and it will stay healthy longer than any micronic filter you know of.

Handling a wide range of fluids at a wide range of flow rates . . . the MICRO-KLEAN won't *shrink* or *swell* or *channel*. So there's no place for the fluid to go except where it's supposed to . . . through the cartridge.

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Check the box nearby for an explanation of MICRO-KLEAN's exclusive performance.

No Fluid Is Better Than Its Filtration

EXPLANATION

Why won't MICRO-KLEAN *shrink or swell or channel?*
Because the fiber structure is resinous-impregnated and polymerized, and resists any deteriorating effects of most fluids and contaminants.

Why does MICRO-KLEAN *have double dirt capacity?*
Because exclusive "graded density in depth" permits smaller particles to penetrate to varying depths—no surface-loading.

Why does MICRO-KLEAN *take up less space?*
Because the cartridge is *all filter*—no structural elements, no cans, bags, springs, inserts. Full-flow filtration needs much less room.

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